TRANSACTIONS

American Society of Heating and Ventilating Engineers

VOLUME 53

Fifty-Third Annual Meeting
Cleveland, Ohio
JANUARY 27-31, 1947

Semi-Annual Meeting
Coronado, Calif.
JUNE 1-4, 1947

PUBLISHED BY THE SOCIETY AT THE OFFICE OF THE SECRETARY
51 Madison Avenue
NEW YORK 10, N. Y.
PRINTED IN U.S.A.

Printed and bound by
The Horn-Shafer Co.
Baltimore, Md.
U.S.A.

CONTENTS

Officers and Council—1947

Headquarters

51 Madison Ave., New York 10, N. Y.
(Tel.: MUrray Hill 3-0291)

Officers

President .... BALDWIN M. WOODS
First Vice President .... GEORGE L. TUVE
Second Vice President .... ALFRED E. STACEY, JR.
Treasurer .... JOHN F. COLLINS, JR.
Secretary .... A. V. HUTCHINSON
Technical Secretary .... CARL H. FLINK
Assistant to President .... CLYDE A. MCKEEMAN

Council

BALDWIN M. WOODS, *Chairman* GEORGE L. TUVE, *Vice Chairman*

Three Years	*Two Years*	*One Year*
M. W. BISHOP	E. G. CARRIER	W. A. DANIELSON E. N. MCDONNELL
CARL F. BOESTER	F. W. HUTCHINSON	A. J. OFFNER H. R. ROTH
LEO HUNGERFORD	R. A. SHERMAN	ERNEST SZEKELY
R. F. TAYLOR	M. S. WUNDERLICH	L. P. SAUNDERS, *Ex-Officio*

Council Committees

Executive—BALDWIN M. WOODS, *Chairman;* A. E. STACEY, JR., G. L. TUVE, J. F. COLLINS, JR., *Ex-Officio.*

Finance—A. E. STACEY, JR., *Chairman;* R. A. SHERMAN, ERNEST SZEKELY, J. F. COLLINS, JR., *Ex-Officio.*

Meetings—G. L. TUVE, *Chairman;* E. G. CARRIER, LEO HUNGERFORD.

Membership—M. W. BISHOP, *Chairman;* F. W. HUTCHINSON, H. R. ROTH.

Standards—M. S. WUNDERLICH, *Chairman;* CARL F. BOESTER, W. A. DANIELSON.

Advisory Council

ALFRED J. OFFNER, *Chairman;* HOMER ADDAMS, M. F. BLANKIN, W. H. CARRIER, S. E. DIBBLE, S. H. DOWNS, W. H. DRISCOLL, E. O. EASTWOOD, W. L. FLEISHER, H. P. GANT, F. E. GIESECKE, E. HOLT GURNEY, L. A. HARDING, H. M. HART, C. V. HAYNES, E. VERNON HILL, JOHN HOWATT, D. D. KIMBALL, G. L. LARSON, S. R. LEWIS, J. F. MCINTIRE, F. B. ROWLEY, A. C. WILLARD and C.-E. A. WINSLOW.

Special Committees

Admission and Advancement: E. L. CROSBY, *Chairman;* E. R. QUEER (*two years*), T. F. ROCKWELL (*three years*).

A.S.H.V.E. *Code for Testing Heavy Duty Furnaces:* E. K. CAMPBELL, *Chairman;* BOWEN CAMPBELL, R. S. DILL, F. A. KITCHEN, A. P. KRATZ, W. J. MAGIRL, B. F. MCLOUTH, F. L. MEYER, F. J. NUNLIST, A. A. OLSON, B. B. REILLY, DR. H. J. ROSE, H. A. SOPER.

Business and Financial Policies: E. N. MCDONNELL, *Chairman;* M. F. BLANKIN, JOHN F. COLLINS, JR., L. E. SEELEY, A. E. STACEY, JR.

Special Committees *(continued)*

Chapter Relations: R. F. Taylor, *Chairman;* F. T. Ball, W. G. Boales, C. S. Koehler, J. Donald Kroeker, F. A. Leser, H. King McCain, G. W. F. Myers, D. L. Taze.

Constitution and By-Laws: S. H. Downs, *Chairman;* John James, H. E. Sproull.

F. Paul Anderson Award: G. L. Tuve, *Chairman;* C. M. Ashley, F. E. Giesecke, F. C. McIntosh, D. W. Nelson.

Guide Publication: R. S. Dill, *Chairman;* R. L. Byers, R. P. Cook, R. C. Cross, B. H. Jennings, C. F. Kayan, W. M. Wallace, II, W. N. Witheridge, C. P. Yaglou, Cyril Tasker, *Ex-Officio.*

History: John F. Collins, Jr., *Chairman;* Homer Addams, E. K. Campbell, W. H. Carrier, W. H. Driscoll, W. L. Fleisher, S. R. Lewis, F. C. McIntosh, G. D. Winans.

Nominating: Neil H. Peterson, San Francisco, Calif., *Chairman;* J. E. Haines, Minneapolis, Minn., *Secretary;* H. E. Adams, South Norwalk, Conn.; A. W. Edwards, Cincinnati, Ohio; C. Rollins Gardner, Dallas, Tex.; John James, Cleveland, Ohio; L. F. Kent, Atlanta, Ga.; A. S. Morgan, Toronto, Ont., Canada; C. H. Pesterfield, East Lansing, Mich.; J. R. Vernon, Milwaukee, Wis.; E. C. Willey, Corvallis, Ore. *Alternates:* Leo Garneau, Montreal, Que., Canada, and L. T. Mart, Kansas City, Mo.

Promotion of Research: J. E. Haines, *Chairman;* A. J. Hess, J. B. Hewett, F. A. Leser, C. E. Price.

Publication: J. C. Fitts, *Chairman;* John A. Goff (*two years*), John James (*three years*).

To Cooperate with Other Professional Societies: G. D. Winans, *Chairman;* James Holt, A. P. Kratz.

Chapter Delegates Committee

Chapter	*Delegate*	*Alternate*
Atlanta	H. King McCain	L. L. Barnes
Central New York	E. K. Ormsby, Jr.	C. M. Ashley
Central Ohio	R. B. Breneman	H. R. Allonier
Cincinnati	A. W. Edwards	H. K. Jennings
Connecticut	L. A. Teasdale	A. J. Lawless
Delta	F. G. Burns	J. S. Burke
Golden Gate	C. E. Bentley	John Everetts, Jr.
Illinois	O. J. Prentice	W. A. Kuechenberg
Indiana	I. W. Cotton	Ferdinand Jehle
Iowa	E. O. Olson	C. A. Wheeler
Kansas City	P. C. Leffel	R. B. Mason
Manitoba	Einar Anderson	J. R. Stephenson
Massachusetts	C. W. Larson	P. A. Croney
Memphis	N. C. Ledbetter	J. B. Lammons
Michigan	F. R. Bishop	A. E. Knibb
Minnesota	J. E. Haines	A. B. Algren
Montreal	A. B. Madden	F. A. Hamlet
Nebraska	D. E. McCulley	H. J. Kleinkauf
New York	W. A. Sherbrooke	M. C. Giannini
North Carolina	C. Z. Adams	K. W. Selden, Jr.
North Texas	C. R. Gardner	H. J. Martyn
Northern Ohio	John James	W. R. Moore

Chapter Delegates Committee *(continued)*

Chapter	*Delegate*	*Alternate*
Oklahoma	W. W. Frankfurt	E. G. Spaan, Jr.
Ontario	D. A. Stott	E. G. Spall
Oregon	W. B. Morrison	J. D. Kroeker
Pacific Northwest	E. H. Langdon	W. M. Wallis
Philadelphia	J. O. Kirkbride	L. A. Childs
Pittsburgh	D. W. Loucks	E. H. Riesmeyer, Jr.
Rocky Mountain	J. J. Johnson	H. J. Woehlke
St. Louis	G. W. F. Myers	B. L. Evans
South Texas	C. C. Quin	D. M. Mills
Southern California	Art. Theobald	R. A. Lowe
Southwest Texas	R. W. Barnes	
Utah	E. J. Watts	C. E. Murdock
Virginia	R. C. Thomas	W. H. Webster, Jr.
Washington, D. C.	F. A. Leser	L. B. Nye, Jr.
Western Michigan	C. H. Pesterfield	H. R. Limbacher
Western New York	G. E. Adema	E. J. Woolcock
Wisconsin	F. J. Nunlist	J. R. Vernon

Committee on Research

A.S.H.V.E. Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio
(Tel.: EXpress 6161)

L. P. Saunders, *Chairman*

T. H. Urdahl, *Vice Chairman* A. C. Fieldner, *Ex-Officio*

Cyril Tasker, *Director of Research*

Three Years	*Two Years*	*One Year*
R. C. Cross	L. N. Hunter	R. M. Conner
M. K. Fahnestock	C. O. Mackey	John A. Goff
John James	R. D. Madison	F. W. Hutchinson
F. J. Kurth	L. G. Miller	R. K. Thulman
T. H. Urdahl	L. P. Saunders	W. E. Zieber

Executive Committee

L. P. Saunders, *Chairman*

T. H. Urdahl, *Vice Chairman*

R. M. Conner L. N. Hunter W. E. Zieber

Technical Advisory Committees 1947

Air Cleaning: R. S. Dill, *Chairman;* R. C. Cross,* O. C. Eliason, R. S. Farr, C. D. Graham, A. B. Hubbard, R. E. Hunsaker, D. L. Hunzicker, J. W. May, R. H. Mills, G. W. Penney, R. S. Poole, R. P. Warren, W. N. Witheridge, R. D. Wood.

Air Distribution and Air Friction: Ernest Szekely, *Chairman;* N. E. Berry, S. H. Downs, W. H. Hoppmann, II, F. J. Kurth,* J. N. Livermore, R. D. Madison,* G. E. McElroy, L. G. Miller,* D. W. Nelson, G. B. Priester, C. A. Thorp, T. H. Troller, J. H. Van Alsburg.

* Member of Committee on Research.

Technical Advisory Committees *(continued)*

Air Sterilization: W. F. Wells, *Chairman;* W. T. Anderson, Jr., L. J. Buttolph, C. W. Flood, Jr., Leonard Greenburg, M.D., B. H. Jennings, F. W. Hutchinson,* J. A. Reyniers, Mildred Wells, M.D., C.-E. A. Winslow.

Cooling Load: W. E. Zieber,* *Chairman;* C. M. Ashley, *Vice Chairman;* R. D. Blum, W. F. Friend, R. H. Heilman, H. W. Heisterkamp, R. C. Jordan, C. F. Kayan, J. N. Livermore, R. H. Lock, C. O. Mackey,* C. H. Randolph, J. P. Stewart.

Corrosion: L. F. Collins, *Chairman;* R. M. Conner,* A. S. Gates, Jr., A. E. Griffin, Dr. E. W. Guernsey, Dr. R. L. Hallett, P. G. Kayser, Dr. H. A. Pray, F. N. Speller.

Fuels: E. T. Selig, Jr., *Chairman;* P. R. Achenbach, J. F. Barkley, R. M. Conner,* W. A. Danielson, R. B. Engdahl, L. N. Hunter,* R. C. Johnson, S. Konzo, J. W. Miller, W. M. Myler, Jr., Dr. H. J. Rose, T. H. Smoot, R. K. Thulman,* E. C. Webb.

Glass: R. A. Miller, *Chairman;* A. B. Algren, Dr. F. L. Bishop, Jr., J. E. Frazier, J. S. Herbert, E. H. Hobbie, L. K. Jones, C. O. Mackey,* F. Preston, W. C. Randall, Vic Sanders, H. B. Vincent, G. B. Watkins.

Heating Load: T. F. Rockwell, *Chairman;* C. M. Ashley, R. A. Biggs, A. I. Brown, R. C. Cross,* R. S. Dill, W. S. Harris, J. M. Hartman, S. I. Rottmayer, M. W. Shears, G. H. Tuttle.

Industrial Ventilation: W. N. Witheridge, *Chairman;* A. D. Brandt, W. B. Harris, W. C. L. Hemeon, D. E. Humphrey, J. M. Kane, M. G. Kershaw, L. L. Lewis, L. G. Miller,* H. A. Mosher, Ralph Poole, B. F. Postman, B. R. Small, R. P. Warren, H. E. Ziehl.

Insulation: E. R. Queer, *Chairman;* R. E. Backstrom, C. B. Bradley, H. E. Lewis, L. E. Pasek, H. E. Robinson, Vic Sanders, T. D. Stafford, R. K. Thulman,* T. T. Tucker, P. M. Woodworth.

Panel Heating and Cooling: G. D. Winans, *Chairman;* A. B. Algren, R. L. Byers, P. B. Gordon, L. N. Hunter,* John James,* J. S. Locke, H. F. Randolph.

Group A—Heat Distribution Within and Behind the Panel: L. N. Hunter,* *Chairman;* P. B. Gordon, *Vice Chairman;* A. B. Algren, R. A. Boggs, Wharton Clay, R. S. Dill, H. L. Flodin, F. E. Giesecke, W. S. Harris, C. F. Kayan, G. D. Lain, R. S. Leigh, C. W. Meininger, D. L. Mills, C. W. Nessell, P. S. Park, Jr., S. I. Rottmayer, S. M. Van Kirk, L. H. Yeager.

Group B—Heat Transfer Between the Panel and the Space: John James,* *Chairman;* H. F. Randolph, *Vice Chairman;* C. M. Ashley, J. T. Bergen, C. F. Boester, Linn Helander, A. L. Jaros, Jr., H. A. Lockhart, C. O. Mackey,* R. A. Miller, G. W. Penney, G. G. Sward, J. M. van Nieukerken.

Group D—Controls: J. S. Locke, *Chairman;* A. B. Algren, H. W. Alyea, S. D. Browne, R. L. Campbell, C. M. Garner, P. B. Gordon, W. J. Hajek, A. J. Keating, Dr. Waldo Kliever, H. T. Kucera, R. F. Neess, R. S. Penn, J. K. M. Pryke, E. J. Ritchie, C. W. Signor, N. D. Skinner, A. S. Widdowfield.

Physiological Research: C.-E. A. Winslow, *Chairman;* Thomas Bedford, M.D., A. R. Behnke, M.D., A. C. Burton, M.D., E. F. DuBois, M.D., A. P. Gagge, F. W. Hutchinson,* R. W. Keeton, M.D., D. H. K. Lee, M.D., André Missenard, Charles Sheard.

Sensations of Comfort: C. P. Yaglou, *Chairman;* R. S. Arnold, A. R. Behnke, M.D., Thomas Chester, F. G. Ewens, M. K. Fahnestock,* J. D. Hardy, M.D., F. C. McIntosh, A. B. Newton, K. E. Robinson.

Sorbents: John Everetts, Jr., *Chairman;* F. C. Dehler, John A. Goff,* C. H. B. Hotchkiss, E. R. McLaughlin, J. C. Patterson, G. L. Simpson, F. J. Swaney.

Sound Control: T. A. Walters, *Chairman;* C. M. Ashley, P. H. Geiger, W. H. Hoppmann, II, G. C. Kerr, R. D. Madison,* J. S. Parkinson, A. G. Sutcliffe, T. H. Troller.

---

* Member of Committee on Research.

Officers of Local Chapters–1947

Atlanta

Organized 1937

Headquarters, Atlanta

President.....H. King McCain
Vice-President.....L. L. Barnes
Secretary.....Leo Sudderth, Jr.
Treasurer.....G. R. Bilderback
Board of Governors: M. M. Crout, W. J. McKinney

Central New York

Organized 1944

Headquarters, Syracuse

President.....H. G. Strong
Vice-President.....D. L. Mills
Secretary.....G. F. Keane
Treasurer.....J. H. Carpenter
Board of Governors: J. H. Carpenter, G. F. Keane, R. L. Manier, D. L. Mills, H. K. Ormsby, Jr., H. G. Strong, C. F. Woese

Central Ohio

Organized 1944

Headquarters, Columbus

President.....R. B. Breneman
Vice-President.....W. M. Myler, Jr.
Secretary.....E. A. Norman, Jr.
Treasurer.....A. P. Potts
Board of Governors: H. R. Allonier, A. T. English, N. T. Hess

Cincinnati

Organized 1932

Headquarters, Cincinnati

President.....H. K. Jennings
Vice-President.....M. E. Matthewson
Secretary-Treasurer.....K. B. Little
Board of Governors: A. W. Edwards, H. K. Jennings, K. B. Little, E. W. McNamee, M. E. Matthewson

Connecticut

Organized 1940

Headquarters, New Haven

President.....S. R. Osborne
Vice-President.....E. C. Marsden
Secretary.....Winfield Roeder
Treasurer.....D. M. Hummel
Board of Governors: H. E. Adams, P. D. Bemis, A. J. Lawless

Delta

Organized 1939

Headquarters, New Orleans

President.....J. S. Burke
Vice-President.....L. V. Busenlener
Secretary.....R. B. Guest
Treasurer.....F. G. Burns
Board of Governors: J. S. Adair, Ralph Elizardi, W. P. Oster

Golden Gate

Organized 1937

Headquarters, San Francisco

President.....F. W. Kolb
Vice-President.....H. V. Hickman
Secretary.....K. F. Baldwin, Jr.
Treasurer.....R. C. Cushing
Board of Governors: P. R. Babcock, John Everetts, Jr., James Garner, H. G. Smith

Illinois

Organized 1906

Headquarters, Chicago

President.....W. A. Kuechenberg
Vice-President.....C. M. Burnam, Jr.
Secretary.....G. W. Bornquist
Treasurer.....J. S. Kearney
Board of Governors: C. E. Crone, P. J. Marschall, J. J. Philippi, O. J. Prentice

Indiana

Organized 1943

Headquarters, Indianapolis

President.....I. W. Cotton
Vice-President.....P. R. Jordan
Secretary.....W. R. Fenstermaker
Treasurer.....F. S. Boone
Board of Governors: G. W. Jackson, F. A. Stickle, T. R. Davis

Iowa

Organized 1940

Headquarters, Des Moines

President.....C. A. Wheeler
Vice-President.....D. C. Murphy
Secretary-Treasurer.....D. E. Wells
Board of Governors: W. E. Burch, E. O. Olson, R. S. Stover

OFFICERS OF LOCAL CHAPTERS—1947 *(continued)*

Kansas City

Organized 1917

Headquarters, Kansas City, Mo.

President.............................P. C. Leffel
Vice-President.......................H. E. Gould
Secretary....................Henry Nottberg, Jr.
Treasurer..........................K. M. Stevens
Board of Governors: W. L. Cassell, Carl Clegg, N. W. Downes, S. L. Furber

Manitoba

Organized 1935

Headquarters, Winnipeg

President............................D. S. Swain
Vice-President..................J. R. Stephenson
Secretary-Treasurer..................A. W. Moss
Board of Governors: J. A. Bell, O. J. Hatch, R. E. Moore, A. W. Moss, J. R. Stephenson, D. S. Swain, D. E. Saunders

Massachusetts

Organized 1912

Headquarters, Boston

President......P. A. Croney, Adolph Ehrenzeller*
Vice-President................Adolph Ehrenzeller
Secretary-Treasurer..................R. T. Kern
Board of Governors: E. L. Blair, F. H. Brigham, P. A. Croney, A. Dusossoit, Adolph Ehrenzeller, R. T. Kern, C. M. F. Peterson, W. H. Shipp

Memphis

Organized 1944

Headquarters, Memphis

President........................N. C. Ledbetter
Vice-President.........................E. E. Scott
Secretary..............................A. T. Bevil
Treasurer...........................C. S. Fischer
Board of Governors: A. T. Bevil, R. H. Hoshall, T. O. S. Humphrey, Jr., J. B. Lammons, N. C. Ledbetter, E. E. Scott, W. E. Thorpe

Michigan

Organized 1916

Headquarters, Detroit

President............................A. E. Knibb
Vice-President........................E. F. Hyde
Secretary............................L. A. Burch
Treasurer.........................C. F. Donohoe
Board of Governors: G. W. Akers, F. R. Bishop, T. H. Mabley, R. H. Oberschulte

Minnesota

Organized 1918

Headquarters, Minneapolis

President.............................L. C. Gross
Vice-President.........................O. L. Lilja
Secretary.............................L. J. Krause
Treasurer................................E. J. Uhl
Board of Governors: G. M. Brown, J. E. Haines, J. F. Stafford

Montreal

Organized 1936

Headquarters, Montreal, Que.

President.....................T. H. Worthington
Vice-President.......................Leo Garneau
Secretary............................S. W. Salter
Treasurer............................C. A. Booth
Board of Governors: J. J. Cosgrove, J. P. Fitzsimons, F. J. Friedman, A. B. Madden, R. R. Noyes, F. G. Phipps

Nebraska

Organized 1940

Headquarters, Omaha

President.......................Verne Simmonds
Vice-President..................K. R. Magarrell
Secretary..........................G. W. Colburn
Treasurer............................C. A. Carter
Board of Governors: C. A. Carter, G. W. Colburn, H. Y. Kleinkauf, K. R. Magarrell, B. G. Peterson, F. E. Prawl, Verne Simmonds

New York

Organized 1911

Headquarters, New York

President.........................M. C. Giannini
Vice-President....................A. A. Bearman
Secretary...........................Carl H. Flink
Treasurer.........................W. M. Heebner
Board of Governors: P. B. Gordon, Ernst Graber, H. S. Johnson, C. F. Kayan, W. A. Sherbrooke

North Carolina

Organized 1939

Headquarters, Durham

President............................C. Z. Adams
Vice-President......................E. S. DeWitt
Secretary-Treasurer............M. F. DuChateau
Board of Governors: E. H. Brandt, Jr., James Marshall, Arvin Page, L. L. Vaughan

* Filled Unexpired Term.

OFFICERS OF LOCAL CHAPTERS—1947 *(continued)*

North Texas

Organized 1938

Headquarters, Dallas

President....C. R. Gardner
Vice-President....B. S. Foss, Jr.
Secretary-Treasurer....G. A. Linskie
Board of Governors: M. L. Brown, L. S. Gilbert, E. J. Stern

Northern Ohio

Organized 1916

Headquarters, Cleveland

President....John James
Vice-President....W. R. Moore
Secretary....W. M. Rowe
Treasurer....John Richmond
Board of Governors: R. L. Byers, C. M. Humphreys, J. E. Wilhelm

Oklahoma

Organized 1935

Headquarters, Oklahoma City

President....W. W. Frankfurt
Vice-President....W. M. Carroll
Secretary-Treasurer....J. H. Spaan, Jr.
Board of Governors: E. F. Dawson, G. T. Donceel, C. Y. Trowbridge

Ontario

Organized 1922

Headquarters, Toronto

President....D. A. Stott
Vice-President....E. G. Spall
Secretary-Treasurer....H. R. Roth
Board of Governors: J. H. Fox, V. J. Jenkinson, William Philip, A. J. Strain

Oregon

Organized 1939

Headquarters, Portland

President....C. W. Brissenden
Vice-President....E. R. Lokey
Secretary....L. G. Williams
Treasurer....R. C. Chewning
Board of Governors: W. J. Kollas, W. R. Norte, W. B. Morrison

Pacific Northwest

Organized 1928

Headquarters, Seattle, Wash.

President....E. H. Langdon
Vice-President....L. L. Bysom
Secretary....E. J. Rosen
Treasurer....J. D. Sparks
Board of Governors: M. N. Musgrave, R. E. LeRiche

Philadelphia

Organized 1916

Headquarters, Philadelphia

President....E. H. Dafter
1st Vice-President....J. O. Kirkbride
2nd Vice-President....F. H. Buzzard
Secretary....L. A. Childs
Treasurer....J. W. McElgin
Board of Governors: F. H. Buzzard, L. A. Childs, E. H. Dafter, M. G. Kershaw, J. O. Kirkbride, J. W. McElgin, E. K. Wagner

Pittsburgh

Organized 1919

Headquarters, Pittsburgh

President....D. W. Loucks
Vice-President....H. E. Park
Secretary....E. H. Riesmeyer, Jr.
Treasurer....B. B. Reilly
Board of Governors: E. C. Hach, A. F. Nass, C. H. Schneider

Rocky Mountain

Organized 1944

Headquarters, Denver, Colo.

President....J. F. Mohan
Vice-President....B. A. Brickham
Secretary....H. C. Kugeler
Treasurer....C. B. Hickey
Board of Governors: F. L. Adams, F. C. Allen, F. L. Trautman

St. Louis

Organized 1918

Headquarters, St. Louis

President....J. H. Carter
1st Vice-President....B. L. Evans
2nd Vice-President....J. A. Russell
Secretary....J. S. Rosebrough
Treasurer....C. H. Burnap
Board of Governors: L. L. Hamig, B. C. Simons, W. D. Thompson, Ralph Toensfeldt

South Texas

Organized 1938

Headquarters, Houston

President.....D. M. Mills
Vice-President.....C. C. Quin, Jr.
Secretary.....L. L. Ladewig
Treasurer.....R. J. Salinger
Board of Governors: C. L. Boehler, F. C. Brandt, B. P. Fisher

Southern California

Organized 1930

Headquarters, Los Angeles

President.....R. A. Lowe
Vice-President.....R. S. Farr
Secretary.....J. L. Blake
Treasurer.....L. J. Helms
Board of Governors: L. B. Davenport, J. L. McCullough, Nicholas Nassir, Arthur Theobald

Southwest Texas

Organized 1946

Headquarters, San Antonio

President.....F. C. Benham, Jr.
Vice-President.....G. R. Rhine
Secretary-Treasurer.....L. S. Pawkett
Board of Governors: R. W. Barnes, F. C. Benham, Jr., D. E. Locher, A. J. Rummel, E. P. Weatherby, Jr., I. W. Wilke

Utah

Organized 1944

Headquarters, Salt Lake City

President.....E. V. Gritton
Vice-President.....E. J. Watts
Secretary-Treasurer.....C. E. Ferguson
Board of Governors: D. B. Holford, H. G. Richardson, Alfred Richeda, J. T. Young, Jr.

Virginia

Organized 1946

Headquarters, Norfolk

President.....R. C. Thomas
Vice-President.....W. H. Webster, Jr.
Secretary.....J. F. Reynolds
Treasurer.....W. G. Hayes
Board of Governors: W. P. Robinson, R. C. Thomas, J. E. White

Washington, D. C.

Organized 1935

Headquarters, Washington, D. C.

President.....H. H. Hill
Vice-President.....A. S. Gates, Jr.
Secretary.....J. G. Muirhead
Treasurer.....P. R. Achenbach
Board of Governors: J. C. Benson, A. C. Crawford, L. Bert Nye, Jr.

Western Michigan

Organized 1931

Headquarters, Grand Rapids

President.....Frank Harbin, Jr.
Vice-President.....H. W. Wolters
Secretary.....W. C. DeRoo
Treasurer.....C. A. Simonds, L. A. Calcaterra*
Board of Governors: H. R. Limbacher, J. W. Miller, K. E. Robinson

Western New York

Organized 1919

Headquarters, Buffalo

President.....G. E. Adema
1st Vice-President.....Edwin Woolcock
2nd Vice-President.....F. J. Weber
Secretary.....J. H. Bryce
Treasurer.....B. C. Candee
Board of Governors: M. C. Beman, Joseph Davis, W. R. Heath, Herman Seelbach, Jr.

Wisconsin

Organized 1922

Headquarters, Milwaukee

President.....M. W. Bishop, J. R. Vernon*
Vice-President.....W. A. Ouweneel
Secretary.....B. M. Kluge
Treasurer.....W. H. Stevens
Board of Governors: E. W. Gifford, I. J. Haus, H. W. Schreiber

Student Branch

Organized 1946

Headquarters, Texas A. M. College
College Station, Tex.

President.....G. H. Jackson
Vice-President.....S. E. Ammons
Secretary-Treasurer.....T. V. Burns, Jr.
Board of Governors: J. S. Hopper, W. E. Long, W. E. Rohrabacher

* Filled Unexpired Term.

1304

No. 1304

FIFTY-THIRD ANNUAL MEETING, 1947

CLEVELAND, OHIO

THE 53rd annual meeting of the Society attracted an attendance of 2165 members and guests—the largest number ever present at a Society meeting. The new A.S.H.V.E. Research Laboratory at 7218 Euclid Ave., Cleveland, which was purchased by the Society in 1946, was dedicated and was open for inspection during the entire annual meeting, thus affording the members an opportunity to become acquainted with the laboratory staff and to see the facilities available for the contemplated research program.

FIRST SESSION—MONDAY, JANUARY 27, 10:00 A.M.

The first session was held in the new Research Laboratory in order to include the dedication ceremonies and was attended by 500 members and guests. J. E. Wilhelm, Cleveland, president of Northern Ohio Chapter, welcomed the members to the meeting. Pres. Alfred J. Offner, New York, extended the thanks of the Society to the Northern Ohio Chapter for their hearty welcome.

President Offner then introduced Dr. Arthur C. Willard, Urbana, Ill., past president of the Society and president emeritus of the University of Illinois, who gave the address of dedication as follows:

This is a notable occasion in many respects in the history of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

We have met here to dedicate a building devoted to research in those arts which most intimately affect the comfort and health of human beings.

This enterprise is not new, as it already has an impressive record of achievement preserved in the many volumes of TRANSACTIONS published by the Society. The achievements have been made possible by the efforts of many men and they will have an inspiring influence on the staff who will carry on here the search for new and better scientific knowledge in the many fields related to human comfort and health.

It is certainly appropriate at this time to review briefly the origin and development of the Research Laboratory which has a history extending back some thirty years. In so doing, we will pay tribute to the *founding fathers* who conceived the idea of

such a Laboratory. Such recognition is due these far-seeing and courageous pioneers in research and it is good for us who are here today to recall them and their hopes and aspiration.

I think you will all be interested in the first formal action taken in 1917 by the Society, during the presidency of J. I. Lyle, on the establishment of a Research Laboratory. The Record appears in the TRANSACTIONS for 1917, Vol. XXIII, p. 322 and following. Here is part of what you will find:

REPORT OF COMMITTEE* ON RESEARCH BUREAU

At the last Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the following resolution was adopted.

RESOLVED: That a committee be appointed to investigate the possibility of establishing a Research Laboratory on some basis and the ways and means for starting it; one member of this Committee is to be from the Council.

Our President, in pursuance with the action above referred to, appointed a committee which at this time desired to submit for consideration the following:

FIRST: Your committee believes that the time has arrived when the Society needs a Research Laboratory;

SECOND: Ways and means of establishing such laboratory considered by your committee were as follows:

(*Note*: Items a, b, c, d, which follow refer to details of *ways and means* and are omitted because of space and time limitation.)

In conclusion, your committee would recommend the establishment of a Research Laboratory, preferably by the U. S. Bureau of Standards, under the control of the Society; if that is not feasible, to establish a Chair at a University whose students shall make tests under the direction of an authorized member of the Society, and that the expense of the establishment of such a Chair shall be met by an increase in the dues, and the establishment of life members as above outlined.

Your Committee respectfully submits the foregoing as its report, and would ask that it be discharged.

COMMITTEE ON RESEARCH BUREAU

GEORGE W. BARR, *Chairman.*

H. M. HART — W. W. MACON
J. D. HOFFMAN — JAMES W. H. MYRICK
WM. F. MCDONALD — PERRY WEST

On motion of Frank K. Chew, the Report was received and the proposed increase in dues for research was immediately discussed and the Committee was continued.

The TRANSACTIONS for 1918—F. R. Still, president, and for 1919—W. S. Timmis, president, contain very complete reports on the proposal to establish a Research Laboratory under the sole control of the Society. Throughout this period, the whole membership enthusiastically supported the proposal and this support was especially marked by the devoted work of Homer Addams, J. R. Allen, F. F. Bahnson, C. A. Booth, G. A. Chaffee, F. K. Chew, J. H. Davis, S. E. Dibble, M. W. Franklin, F. E. Giesecke, H. M. Hart, J. D. Hoffman, T. F. Humphreys, A. S. Kellogg, G. B. Larimer, E. E. McNair, L. W. Moon, E. T. Murphy, C. W. Obert, Arthur Ritter, and W. S. Timmis.

On August 1, 1919, during the presidency of Walter S. Timmis, The Research Laboratory was officially established and began to function at the United States Bureau of Mines, Pittsburgh, Pa.

The first Director of the Laboratory was John R. Allen, a brilliant scientist who had served as Dean of Mechanical Engineering, University of Michigan and as Dean of Engineering and Architecture at the University of Minnesota.

---

* This committee was appointed pursuant to a resolution passed at the Annual Meeting of the Society in 1916 during the presidency of H. M. Hart.

After the tragic death of Dean Allen in October 1920, Dean L. A. Scipio of Robert College, Constantinople, Turkey, served for a year during his leave of absence from the College.

On August 1, 1921, the directorship of the Laboratory was assumed by F. Paul Anderson, Dean of the College of Engineering, University of Kentucky.

In September 1925, Ferry C. Houghten became the Director and served until his entry into the Navy in World War II.

On October 1, 1943, Cyril Tasker was appointed Director of Research and took charge of the new Laboratory facilities at Cleveland, which were temporarily located at 10700 Euclid Avenue. At the end of the two-year lease, it was found necessary to look for other quarters and a special committee composed of, E. N. McDonnell, *chairman;* M. F. Blankin, L. E. Seeley, A. E. Stacey, Jr., G. L. Tuve, T. H. Urdahl and B. M. Woods, recommended the purchase of a building at 7218 Euclid Ave., which came into the Society's possession in July 1946.

Ever since its establishment, the Research Laboratory has had the loyal and untiring support and cooperation of the members of the Society. They have served on its Research Advisory Committees, and on the numerous and important Technical Advisory Committees. There are now sixteen of these advisory committees with an average membership of 10 men, who are especially qualified to advise in the various research projects under study and investigation by the Laboratory. The present organization of the Committee on Research includes L. P. Saunders who is now chairman. The other members of this Committee are: T. H. Urdahl, Vice Chairman; A. C. Fieldner, Ex-Officio; Cyril Tasker, Director of Research; L. N. Hunter, C. O. Mackey, R. D. Madison, L. G. Miller, R. M. Conner, John A. Goff, F. W. Hutchinson, R. K. Thulman, W. E. Zieber, C. M. Ashley, F. E. Giesecke, F. C. McIntosh, G. L. Tuve. It will, therefore, be apparent that no less than 175 members of the Society are now conversant with and officially responsible for various phases of the over-all research program of the Society.

From their inception, the Research activities of the Society have included the cooperation, through numerous research agreements, of many universities and scientific institutions as well as government agencies. The first of these cooperative research programs was set up in 1919 at the University of Minnesota under the able direction of Prof. F. B. Rowley for the study of radiant heat losses from direct radiators. Other cooperating universities and insitutions include: Carnegie Institute of Technology, Case School of Applied Science (now Case Institute of Technology), Cornell University, University of California, Georgia School of Technology, Oregon State College, University of Pennsylvania, Agricultural and Mechanical College of Texas, Lehigh University, Rensselaer Polytechnic Institute, University of Illinois, and the University of Wisconsin.

As an important section of this brief review of the origin and development of the Research Laboratory during the past thirty years, I wish to pay special tribute and grateful acknowledgment for the invaluable financial support received through the generous contributions of funds from industrial firms and various associations and institutes. Again, limitations of space and time prevent me from naming each of these contributors but a fairly complete list of these donors appears in the TRANSACTIONS for 1944, Vol. 50, p. 30, under the Report of the Committee on Research, Chairman C. M. Ashley, covering *Twenty-five Years of Research.*

The amount of these sustaining contributions over the years is most impressive, since they total more than half of all the expenditures for Society research from 1919 to 1946, inclusive. That total is approximately $925,000—of which $445,000 came from Society dues and appropriations while approximately $480,000 came from industry, association, expositions, the U. S. Navy, individuals and other sources. In this connection, it should be noted that in 1925 the Society dues were raised to $25 and 40 percent of all dues were allocated to the Research Fund.

At this point it would be appropriate, if time permitted, to introduce by title, at least, all of the many research projects sponsored by the Society, for which time, thought, energy and money have been spent since the Laboratory was inaugurated at the U. S. Bureau of Mines in 1919. The list of these projects would be most impressive in both quality and quantity as the number of papers and reports resulting from research at the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory and at cooperating institutions is approximately 285. This total does not include literally hundreds of articles, reviews, summaries and interpretations of Society research which have appeared in various technical and trade journals and in the scientific journals of other professional societies. The Research Laboratory and the Society owe a great debt to these other publications for disseminating and publicizing the results of these researches. Research, such as ours, which is conducted largely in the public interest can only become really effective when it is given the widest possible publicity and adequately interpreted. For this purpose, scientific journals alone are not adequate and never will be.

This vast fund of knowledge based on the sciences related to the arts of heating, ventilating, and air conditioning is all recorded in the TRANSACTIONS of the Society for the benefit of all mankind. The members of the Society are justifiably proud of such a record and the further fact that this Society is the only professional engineering organization which maintains and operates its own Research Laboratory.

The results of this research must be interpreted and translated by industry into the processes and equipment necessary to improve the physical environment of our homes, offices, hotels, theaters, churches, and factories, as well as our trains, ships, and planes for greater human comfort, health, happiness, and efficiency. No field of research demands more skill in the successful application of its scientific findings than the field of heating, ventilating, and air conditioning. This Society must never forget that its research, no matter how brilliant the staff and their achievements, is not an end in itself. Only the trained engineer can make the results of scientific research in this field fully effective for the benefit of mankind.

And so with this wonderful record of achievement to inspire us all, we dedicate this new home of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS to the discovery of new knowledge and the improvement and extension of our present knowledge in all fields related to the arts of heating, ventilating, and air conditioning. *This Laboratory will always seek to promote the comfort and health of human beings through scientific research.*

I congratulate President Offner and Director Tasker and his staff as we face the future and *carry on* in the best research traditions of our Society for service to mankind.

President Offner thanked Dr. Willard for dedicating the Laboratory to the service of the membership and the heating and ventilating profession.

G. Brooks Earnest, president of The Cleveland Technical Societies Council, expressed his pleasure at the establishment in Cleveland of the Society's Research Laboratory, which he felt was a valuable addition to the facilities already established in Cleveland for engineering development.

Dr. William E. Wickenden, president of Case School of Applied Science (now Case Institute of Technology), Cleveland, referred to the close cooperation that had existed between Case and the Society in promoting research in problems of interest to both. He pointed out that the cooperative research work had been of value, not only to the Society, but even more so to Case in providing an opportunity for development of engineers who were thereby fitted to render valuable service to the industry.

President Offner then presented the president's report.

PRESIDENT'S REPORT

The year 1946, just passed, was an eventful year for our Society in many ways. It has moved forward in research, membership, publications and in its value to the public, industry and its members.

During the past year we have consummated our hopes and aspirations of many years—to own our own Laboratory property. The purchase of the buildings and property located on Euclid Avenue in Cleveland was the largest financial transaction ever undertaken by the Society. It represents an investment of over $80,000—$70,000 for the property and over $10,000 for improvements. One half of the purchase price, or $35,000 was paid on delivery of the deed, the balance being covered by a bank mortgage. Part of the principal has already been paid off. We have hopes of liquidating the remaining balance of the mortgage in the very near future by means of voluntary contributions from members and friends of the Society.

The purchase of the Laboratory property, our increased research program, increases in the costs of publishing THE GUIDE and the increase in the costs of other activities of the Society have made large demands on our finances. However, prudent management and care leave our financial position at the end of our fiscal year in a sound position.

The Fund Raising Campaign for Research from industry was inaugurated during the past year. These funds are to supplement the 40 percent allocated to research from membership dues, as required by our Constitution, the moneys allocated by the Council from time to time and the profits received from our editorial contract, THE GUIDE and from the Heating and Ventilating Exposition. The funds are being solicited from industry and their trade associations on a business basis to carry on basic studies of technical problems which are of benefit to industry. Therefore, these funds are not solicited as a contribution or charity but for actual value received. The Society has engaged the services of Clyde A. McKeeman, a member, to actively carry on this work and he has been laying the groundwork for this very important work during the past few months. We have great hopes and expectations for this new endeavor of the Society.

Our Research Laboratory is unique among engineering societies, in that we are the only national engineering Society to own and operate such facilities. It deserves the full support, not only of industry, but of every member of the Society.

The 1947 issue of THE GUIDE will be the 25th Anniversary edition. During the 25 years of its existence, it has grown both in importance, in usefulness, in the amount of technical and usable information and in size. It has grown from an issue of 5000 copies in 1922 to over 17,000 copies for the 1947 issue, from 192 pages of technical data to 912 pages, and from 133 pages of manufacturers information to 334 pages for the new issue. The 12,000 copies of the 1946 issue of THE GUIDE that were printed were completely sold out, necessitating a second printing of 3000 copies which were sold before they came off the press.

A change in our policy of publishing research papers has been made. In the future, research papers may be published in the JOURNAL, either as a complete report or in abstract form. If published in abstract form, the complete report will be published as a separate bulletin for distribution to members and others interested in same.

A change in the method of nominating the officers and members of the Council, as voted by the membership, goes into effect at this meeting. The new nominating committee consists of 11 members and 2 alternates, distributed in 7 geographic areas, 4 selected by the Council and 7 by the Chapter Delegates Committee. This committee replaces a much larger nominating committee, consisting of a member from each Chapter; which method has been in use for many years.

One of the duties imposed on the president, and shared by the other officers, is the visiting of chapters for the purpose of meeting with and becoming better acquainted with the membership at large. During the three years of such visits, I have traveled from north to south and from east to west, visiting 32 of the 38 chapters we now have. I am sorry that time did not permit my visiting the remaining 6. During the past year, two new chapters have been added, the Virginia Chapter at Norfolk and the Southwest Texas Chapter at San Antonio. A Student Branch has also been formed at Texas A. & M. The Society passed its 5000 membership mark in August and now has nearly 5400 members, an all time high in the Society's history, an increase of about 800 members during the year. This year also finds the resumption of the Heating and Ventilating Exposition after a lapse of seven years due to the war.

I wish to take this opportunity to express to the membership my sincerest thanks for the honor they have conferred upon me and the confidence they have shown in electing me as their President. I hope I have lived up to your expectations. I also wish to thank my fellow officers, the entire Council, all committee chairmen and their members, and all others who have done such splendid work in advancing the interests of the Society. My personal thanks also go to Secretary Hutchinson, Technical Secretary Flink and Director Tasker and all staff members of the New York office and those at Cleveland for their loyal support. Without the help of all these, the Society would not have made the advances it has made during the year 1946. I have enjoyed every minute of my term of office. It also has been a personal privilege and pleasure to serve the Society in all of the 25 years that I have been a member, and I hope I will be permitted to continue to do so in the future.

Respectfully submitted,

ALFRED J. OFFNER, *President*

COUNCIL REPORT

The Council held five meetings since the 52nd Annual Meeting, namely, in New York, St. Louis, Montreal, Atlanta and Cleveland.

At the organization meeting President Offner announced his committee appointments which were unanimously confirmed by the Council. A new office was created, Assistant to the President, and Clyde A. McKeeman was appointed. A. V. Hutchinson was reappointed Secretary; Carl H. Flink was continued as Technical Secretary. Depositories for funds were selected and a certified public accountant was appointed.

The Council voted to purchase property in Cleveland for the Research Laboratory, at a cost of $70,000 for the land and buildings at 7218 Euclid Ave. For the down payment of $35,000 current funds were used with the amount in excess of $50,000 in the Reserve Fund. A proposal was made that money from the present Endowment Fund be obtained to replenish the operating funds used. An additional $10,000 was allocated for moving and alteration expenses.

With the new Charter in effect a study of the Constitution and By-Laws was authorized. An increase in the advertising rates for THE GUIDE, 1947, was put into effect.

Meeting places for June 1946 and January and June 1947 were selected.

Petitions for a Charter for a Virginia Chapter and a Southwest Texas Chapter were favorably acted upon and a Student Branch at Texas A. & M. College was approved.

An employees' pension plan was studied and approved for professional personnel with a contributory feature.

The Council nominated five men to serve for a three-year term on the Committee on Research and under the new By-Laws, four members and an alternate for the Nominating Committee were selected.

A revised publication procedure and policy was approved.

The Budget for 1946-47 was adopted providing for an estimated income of $170,685 and an expenditure of $158,500 for general activities and $75,000 for Research.

A new form of membership application and some simplification of the election procedure were adopted.

The Council acted in accordance with the Constitution and By-Laws on all matters required, also accepted 39 resignations, canceled the membership of 21 for non-payment of dues, and records with deep regret the death of a Charter Member Charles F. Hauss, former Treasurer F. D. Mensing, 8 Life Members and 29 other members.

Respectfully submitted,

THE COUNCIL

SECRETARY'S REPORT

The year 1946 was an eventful one for the Society and recorded six significant items:

1. The amendment of the Charter.
2. The highest level of membership—5285.
3. The largest number of new member applications in any one year—1029.
4. The acquisition of our own Research Laboratory Building in Cleveland.
5. The largest edition of THE GUIDE ever published and the highest circulation.
6. The organization of two Chapters and one Student Branch.

The Secretary's office handled the innumerable details connected with election of 880 members; the billing and collection of dues, solicitation of Guide advertising and the sale of copies; the editing, proofreading and publishing of meeting papers, Guide text and the TRANSACTIONS, Vol. 51; made arrangements for meetings, officers' trips, chapter speakers, and all other activities required by the Constitution and By-Laws.

With reconversion from war the return of members from service was rapid and reflected by reduction of the service roll from a high of 427 to a present level of 158. Changes of addresses have been very heavy during the year and required much paper work to keep our records clear.

The compiling of Council Minutes, proceedings of meetings, and correspondence with chapters required much time. The Secretary was privileged to visit 15 of the chapters and found them active and progressive in advancing the Society's program of activities.

The records and books of the general and research activities of the Society were kept in the headquarters office, and funds received for deposit were $210,592.70 and disbursements $217,459.21, as verified by the Certified Public Accountant.

Due to a limited staff and several personnel changes during the year, it was not always possible to handle all administrative work as expeditiously as desired, but it is hoped that during this period of rapid Society growth we can effectively keep pace with developments and the needs of the members.

It was a matter of great personal pleasure and satisfaction to me to receive the handsome gift which was presented by President Offner on behalf of the Council and Members in recognition of my 20 years service as Secretary of the Society.

All of the Society's progress in its 52nd year could not have been accomplished without the interest of the members, the unselfish service of the officers and Council, and the cooperation of the various committees. In addition I should like to express my thanks to my associates on the staff, who carried the chief burden of the work and made this a year of record accomplishment.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary*

Dr. Baldwin M. Woods, Berkeley, Calif., first vice president and chairman of the Finance Committee, gave a brief resume of the financial position of the Society and presented the report of the certified public accountant.

ACCOUNTANT'S REPORT

FRANK G. TUSA & CO.
Certified Public Accountants
52 William St., New York

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
51 Madison Ave,
New York, N. Y.

Gentlemen:

Pursuant to your request, we examined the books of account and records of the American Society of Heating and Ventilating Engineers—New York, N. Y., and the related funds for the fiscal year ended October 31, 1946, and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1946. Also, for the fiscal year then ended, we traced the recorded cash receipts into the depositories; we inspected the cancelled bank checks which we compared with the record of cash disbursements; we supported the disbursements by payment vouchers; we accounted for the dues income and interest income from savings accounts and securities.

A Balance Sheet reflecting the financial condition of the Society as of the close of business October 31, 1946, is submitted herewith and your attention is directed to the following comments thereon:

CASH

The Cash on Deposit was verified by direct communication with the commercial and savings banks listed in the schedule of cash included in this report and reconcilement of the amounts reported to us with the respective balances reflected by the books of the Society.

Checks representing the Cash on Hand for deposit were inspected by us and the Petty Cash counted.

MARKETABLE SECURITIES

The securities, shown on the subjoined schedule, were verified by direct communication with the Bankers Trust Company, where same are deposited for safe-keeping. This asset has been included in the Balance Sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon. During 1946 the following securities were sold:

	Cost	Sales Price	Profit or Loss
GENERAL FUND			
10M U. S. Savings Bonds, Series "D"	$ 7,500.00	$ 7,500.00	—0—
ENDOWMENT FUND			
2M U. S. Treasury Bonds 3⅛s—1949	1,941.46	2,000.00	$58.54
1M U. S. Treasury Bonds 3s—1948....	1,024.69	1,000.00	(24.69)
RESERVE FUND			
10M U. S. Savings Bonds, Series "C" of May 1, 1938....	7,500.00	7,500.00	—0—
3M U. S. Treasury Bonds 3s—1948....	3,074.06	3,000.00	(74.06)
Totals....	$21,040.21	$21,000.00	$(40.21)

CERTIFICATE OF INDEBTEDNESS

The certificate of indebtedness issued to the Society was verified by inspection of the instrument and the interest accounted for.

ACCOUNTS RECEIVABLE

A trial balance of the membership dues receivable taken as of the close of business October 31, 1946, was classified as to membership and aged as follows:

CLASSIFICATION	Amount
Members	$3,798.00
Associates	3,528.75
Juniors	346.00
Students	6.00
Total	$7,678.75

AGING	Amount
Dues Invoiced during 1946	$6,436.25
Dues Invoiced during 1945	864.50
Dues Invoiced prior years	378.00
Total	$7,678.75

The Miscellaneous Accounts receivable were classified and aged as follows:

CLASSIFICATION	Amount
Guides (Advertising and Copy Sales)	$ 9,222.11
Transactions	94.00
Books and Reprints	558.42
Art and Engravings	68.83
Sundry	185.29
Total	$10,128.65

AGING	Amount
Charges made during October, 1946	$ 8,447.82
Charges made during September and August, 1946	368.28
Charges made during January to June, 1946	918.20
Charges made prior	394.35
Total	$10,128.65

The reserves for doubtful dues and miscellaneous accounts receivable, in our opinion are ample to cover probable losses that might result during realization.

INVENTORIES

The emblems and TRANSACTIONS on hand on October 31, 1946, were counted by us. THE GUIDE paper was verified by direct communication with the printers. All inventories were priced and computed by us. The inventory of TRANSACTIONS follows:

Volume	Year	Quantity	Price	Amount
Prior	Prior	1350	$1.00	$1,350.00
46	1940	23	1.25	28.75
47	1941	2	1.32	2.64
48	1942	103	1.42	146.26
49	1943	206	1.39	286.34
50	1944	191	1.82	347.62
			Total	$2,161.61

EXCHANGES

This account contains amounts due from Council Members for air travel accommodations which were reimbursed prior to the close of this audit.

MORTGAGES RECEIVABLE

The Society holds a mortgage dated November 15, 1945, to which Cyril and Ada R. Tasker are jointly liable to a balance of $7,449.73. This indenture covers premises 3538 Edison Road, County of Cuyahoga, Cleveland Heights, Ohio, is payable in monthly installments of not less than $81.04, including interest and matures on or before November 15, 1955.

PERMANENT ASSETS

On February 21, 1946, the Society purchased from the Order Sons of Italy of America, Grand Lodge of Ohio, premises located at 7218 Euclid Avenue, Cleveland, Ohio, for the sum of $70,000.00. It received a warranty deed to this property which was recorded in Cuyahoga County, Volume 6050, Page 676 on March 21, 1946, No. 3,149,607. This asset has been appraised as follows:

	Value	Per Cent
Land	$ 24,000.00	18%
Building	113,000.00	82%
Totals	$137,000.00	100%

Depreciation of the building and improvements thereon has been provided for at the annual rate of 2 percent and furniture and fixtures at 10 percent.

DEFERRED CHARGES

We have deferred to future income one-sixth of all the subscriptions paid to *HPAC* since the payment of same is on a calendar year basis and the fiscal year of the Society ends on October 31st, and also, promotion of copy sales applicable to THE GUIDE 1947.

ACCOUNTS PAYABLE

All purchase invoices found on file that were applicable to the operations of the current fiscal period were listed and the proper liability therefore reflected in the attached Balance Sheet.

TAXES WITHHELD

The sum of $423.80 represents Federal Income Taxes withheld from salaries paid to employees during the month of October, 1946.

DEFERRED INCOME

On October 31, 1946, the prepaid dues by elected members were:

Members	$ 725.34
Associates	463.25
Juniors	44.00
Students	15.00
Total	$1,247.59

In addition there were dues in the sum of $467.00 prepaid by members proposed for membership.

RESERVE FOR PUBLICATIONS

In accordance with the provisions made in the 1946 budget we have reserved the sum of $5,500.00 to cover the publication of TRANSACTIONS, Vol. 52.

MORTGAGE PAYABLE

Premises located at 7218 Euclid Avenue, Cleveland, Ohio, were purchased subject to a mortgage of $35,000.00 dated February 11, 1946, and payable to the Cleveland Trust Company in quarterly installments of $875.00 each beginning with June 1, 1946. This indenture bears interest at the rate

of 4½ percent per annum payable on each installment date and matures on February 11, 1956. Our verification of this liability was made by direct communication.

Up to October 31, 1946, the following contributions were received which are to be applied towards the reduction of the above mortgage at the direction of the council:

New York Chapter	$634.70
Pittsburgh Chapter	200.00
Heating, Piping and Air Conditioning Contractors National Association	100.00
Total	$934.70

FUNDS

An analysis reflecting the changes that occurred in the following Funds during the fiscal year ended October 31, 1946, is subjoined:

General Fund
Reserve Fund
Endowment Fund
F. Paul Anderson Fund
Mortgage Reduction Fund

INCOME AND EXPENSES

A statement in summary form reflecting the Income and Expenses and comparable items budgeted for the fiscal year ended October 31, 1946, of the Society GUIDE and Research follows:

	Actual	Budget
SOCIETY		
Income	$76,024.04	$73,893.00
Expenses	79,618.04	80,425.00
NET OUT-GO	(3,594.00)	(6,532.00)
GUIDE		
Income	85,646.17	59,400.00
Expenses	69,545.60	53,725.00
NET INCOME	16,100.57	5,675.00
RESEARCH		
Income	48,922.49	107,460.00
Expenses	60,702.12	92,870.00
NET INCOME OR OUT-GO	(11,779.63)	14,590.00
TOTAL NET INCOME	$ 726.94	$13,733.00
CAPITAL ITEMS		
New Equipment for Research	$ 4,828.60	$ 7,000.00
Initiation Fees to Reserve Fund	7,593.45	4,800.00
Real Estate and Improvements (Cash Disbursed)	49,527.27	—0—
TOTAL	$61,949.32	$11,800.00

Respectfully submitted,

Frank Tusa & Co.
Certified Public Accountants.

Dated January 9, 1947.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.
October 31, 1946

ASSETS

GENERAL FUND				
CASH				
On Deposit		$20,786.24		
On Hand for Deposit		5.00		
On Hand		100.00	$20,891.24	
INVESTMENTS (AT COST)				
Securities (Market Value $15,495.00)		15,400.00		
Add: Accumulated Interest	$ 270.00			
Add: Accrued Interest	105.00	375.00	15,775.00	
CERTIFICATE OF INDEBTEDNESS			75.52	
ACCOUNTS RECEIVABLE				
Membership Dues		7,678.75		
Less: 40% for Research	2,930.70			
Less: Reserve for Doubtful	184.13	3,114.83		
		4,563.92		
Advertising and Sundry Debtors	10,128.65			
Less: Reserve for Doubtful	112.50	10,016.15	14,580.07	
INVENTORIES				
TRANSACTIONS		2,161.61		
Emblems		295.31		
GUIDE Paper		7,601.91	10,058.83	
EXCHANGES			1,010.68	
MORTGAGE RECEIVABLE			7,449.72	
PERMANENT				
Land and Building	70,120.80			
Improvements	12,656.59			
	82,777.39			
Less: Reserve for Depreciation	1,036.66	81,740.73		
Library		300.00		
Furniture and Fixtures	2,941.99			
Less: Reserve for Depreciation	1,696.41	1,245.58	83,286.31	
DEFERRED CHARGES				
Prepaid *HPAC* Subscriptions		1,534.72		
1947 GUIDE Copy Sales Promotion		1,243.21	2,777.93	$155,905.30
RESERVE FUND				
CASH				
On Deposit		6,635.27		
On Hand for Deposit		906.23	7,541.50	
Securities at Cost (Market Value $42,458.50)		40,145.00		
Add: Accumulated Interest		2,313.50	42,458.50	50,000.00

ENDOWMENT FUND				
CASH				
On Deposit			5,943.42	
Securities at Cost (Market Value $20,846.76)		19,039.50		
Add: Accumulated Interest	1,549.76			
Add: Accrued Interest	20.00	1,569.76	20,609.26	26,552.68
F. PAUL ANDERSON FUND				
CASH				
On Deposit		21.68		
On Hand for Deposit		12.50	34.18	
Securities at Cost (Market Value $978.00)		1,000.00		
Add: Accrued Interest		12.50	1,012.50	1,046.68
MORTGAGE REDUCTION FUND				
CASH				
On Hand for Deposit				934.70
				$234.439.36

LIABILITIES AND FUNDS

GENERAL FUND				
LIABILITIES				
ACCOUNTS PAYABLE			$20,081.30	
FEDERAL WITHHOLDING TAXES			423.80	
ACCRUED ACCOUNTS				
Additional Compensation—Employees			6,074.93	
DEFERRED INCOME				
Prepaid Membership Dues	$1,247.59			
Less: 40% Prepaid to Research	475.44	772.15		
Dues Prepaid by Candidates for Membership		467.00	1,239.15	
RESERVE FOR PUBLICATION				
TRANSACTIONS (1945), Volume 51		251.24		
TRANSACTIONS (1946), Volume 52		5,500.00	5,751.24	
MORTGAGE PAYABLE				
Cleveland Trust Company			33,250.00	
TOTAL LIABILITIES			66,820.42	
GENERAL FUND			89,084.88	$155,905.30

Note "A"—This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

RESERVE FUND				
Principal				$ 50,000.00
ENDOWMENT FUND				
Principal			26,129.20	
Expended Income			423.48	26,552.68

F. PAUL ANDERSON FUND		
Principal	1,033.98	
Unexpended Income	12.70	1,046.68
MORTGAGE REDUCTION FUND		934.70
		$234,439.36

STATEMENT OF INCOME AND EXPENSES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

For the Fiscal Year Ended October 31, 1946

INCOME

INCOME FROM MEMBERS				
DUES—RENEWALS				
Members and Associates	$68,562.00			
Less: Cancellations	1,199.00	$67,363.00		
Less: 40% to Research Fund		26,945.20	$40,417.80	
Juniors and Students		1,332.00		
Less: Cancellations		33.00	1,299.00	$ 41,716.80
DUES—NEW MEMBERS				
Members and Associates	9,585.00			
Less: 40% to Research Fund		3,834.00	5,751.00	
Juniors and Students			689.50	6,440.50
TOTAL DUES				48,157.30
OTHER INCOME				
Initiation Fees			7,593.45	
Emblems and Certificate Frames			195.44	7,788.89
TOTAL INCOME FROM MEMBERS				55,946.19
INCOME FROM PUBLICATIONS				
Editorial Contract			17,500.00	
GUIDE Sales and Advertisements—Per Schedule Attached			85,646.17	
Sale of TRANSACTIONS			850.56	
Income from Books, Reprints, etc.			505.36	
Servicemen's Fees			134.00	104,636.09
INCOME FROM INVESTMENTS				
Interest—Savings Accounts			95.83	
Interest—Securities			730.72	
Interest—Certificates of Indebtedness			1.26	
Interest—Mortgages			260.12	1,087.93
TOTAL INCOME				$161,670.21

EXPENSES

OPERATING EXPENSES	
President's Fund	$ 2,002.50
Council Travel and Meetings	3,029.86
Membership Committee	1,596.59
Admissions and Advancement Committee	296.06

Constitution and By-Laws Committee	13.67		
Nominating Committee	39.22		
Cardex Books to Chapters	77.69		
Chapter Relations Committee and Records	93.73		
Finance Committee	120.13		
Chapter Delegates Travel	3,234.55		
A.S.A. Membership	100.00		
Medals and Awards	426.72		
Membership Certificates	593.01	11,624.13	
MEETING EXPENSES			
Meetings	2,278.20		
Speakers to Chapters	1,045.32		
Chapter Meeting Allowance	900.00	4,223.52	
PUBLICATION EXPENSES			
Members' Subscriptions, *H.P.A.C.*	9,038.92		
GUIDE Publication and Distribution—Per Schedule Attached	69,545.60		
TRANSACTIONS	8,579.56		
Membership Roll	1,409.00	88,573.08	
HEADQUARTER'S EXPENSE			
Salaries—Secretary and Staff	31,290.89		
Additional Compensation	6,074.93		
Traveling—Secretary and Staff	1,746.34		
Rent and Light	4,006.29		
Telephone	1,233.38		
Telegraph	334.73		
Postage	2,856.82		
General Printing	932.16		
Office Supplies	682.64		
Addressing and Address Changes	263.72		
Professional Services	1,624.20		
Bank Charges and Foreign Exchange	350.21		
Depreciation of Furniture and Fixtures	278.91		
General Office Expense	1,131.11		
TOTAL HEADQUARTER'S EXPENSE	52,806.33		
Less: 30% Applicable to GUIDE	15,841.90	36,964.43	141,385.16
NET INCOME			20,285.05
DEDUCT COUNCIL APPROPRIATIONS			
Initiation Fees to Reserve Fund		7,593.45	
Special Appropriation		237.88	
Research Fund-Raising Campaign		7,540.60	15,371.93
EXCESS OF INCOME OVER EXPENSES FOR THE YEAR			$ 4,913.12

BUDGET COMPARISON—SOCIETY ACTIVITIES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.
For the Fiscal Year Ended October 31, 1946

INCOME

MEMBERSHIP INCOME	Actual	Budgeted	Increases	Decreases
DUES—RENEWALS				
100—Members	$ 24,126.00	$ 24,300.00		$ 174.00
101—Associates	16,291.80	16,740.00		448.20

102—Juniors	1,287.00	1,750.00		463.00
103—Students	12.00	50.00		38.00
	41,716.80	42,840.00		1,123.20
DUES—NEW MEMBERS				
104—Members	2,076.30	1,200.00	876.30	
105—Associates	3,674.70	1,350.00	2,324.70	
106—Juniors	662.50	375.00	287.50	
107—Students	27.00	25.00	2.00	
	6,440.50	2,950.00	3,490.50	
TOTAL DUES INCOME	$ 48,157.30	$ 45,790.00	$ 3,490.50	$ 1,123.20
OTHER INCOME				
108—Initiation Fees	7,593.45	4,800.00	2,793.45	
109-110—Emblems and Certificate Frames	195.44	1,000.00		804.56
	7,788.89	5,800.00	2,793.45	804.56
TOTAL INCOME FROM MEMBERS	55,946.19	51,590.00	6,283.95	1,927.76
INCOME FROM PUBLICATIONS				
115—Editorial Contract	17,500.00	17,250.00	250.00	
116—GUIDE Sales and Advertisements	85,646.17	59,400.00	26,246.17	
117—Sale of TRANSACTIONS	850.56	800.00	50.56	
118—Income from Books, Reprints, etc.	505.36	600.00		94.64
119—Sales Codes	—0—	—0—		
120—Servicemen's Fees	134.00	400.00		266.00
	104,636.09	78,450.00	26,546.73	360.64
INCOME FROM INVESTMENTS				
125—Interest—Savings Accounts	95.83	250.00		154.17
126—Interest—Securities	730.72	1,000.00		269.28
127—Interest—Cert. of Indebt.	1.26	3.00		1.74
Interest—Mortgage	260.12	—0—	260.12	
	1,087.93	1,253.00	260.12	425.19
COLLECTION OF PRIOR YEAR'S DUES	798.64	2,000.00		1,201.36
TOTAL INCOME	$162,468.85	$133,293.00	$33,090.80	$ 3,914.95

EXPENSES

OPERATING EXPENSES				
150—President's Fund	2,002.90	2,500.00		497.10
151—Council Travel to Meetings	3,029.86	2,500.00	529.86	
160—Executive Committee	—0—	100.00		100.00
161—Finance Committee	120.13	200.00		79.87
162—Membership Committee	1,596.59	2,000.00		403.41
170—Admission and Advancement Comm.	296.06	500.00		203.94
171—Constitution and By-Laws	13.67	250.00		236.33
172—Nominating Committee	39.22	100.00		60.78
173B—Chapter Delegates Committee	3,234.55	4,000.00		765.45
173C—Chapter Relations	93.73	250.00		156.27
173D—Cardex Books to Chapters	77.69	100.00		22.31
201—*A.S.A.* Membership	100.00	100.00		

204—Membership Certificates	593.01	200.00	393.01	
206—Medals and Awards	426.72	200.00	226.72	
	11,624.13	13,000.00	1,149.59	2,525.46
Meeting Expenses				
163—Meetings	2,278.20	2,500.00		221.80
173A—Speakers to Chapters	1,045.32	1,500.00		454.68
327—Chapter Meeting Allowance	900.00	900.00		
	4,223.52	4,900.00		676.48
Publication Expenses				
200—Members' Subscriptions, *H.P.A.C.*	9,038.92	9,000.00	38.92	
305-326—Guide Publication and Distribution	69,545.60	53,725.00	15,820.60	
202—Transactions	8,579.56	4,500.00	4,079.56	
203—Membership Roll	1,409.00	1,500.00		91.00
164—Standards (Including Codes)	—0—	150.00		150.00
207—Publication Committee Expenses	—0—	—0—		
	88,573.08	68,875.00	19,939.08	241.00
Headquarter's Expenses				
210—Salaries—Secretary and Staff	31,290.89	30,000.00	1,290.89	
211—Additional Compensation	6,074.93	3,000.00	3,074.93	
212—Traveling—Secretary and Staff	1,746.34	1,500.00	246.34	
213—Rent and Light	4,006.29	4,000.00	6.29	
214—Telephone	1,233.38	800.00	433.38	
215—Telegraph	334.73	350.00		15.27
216—Postage	2,856.82	2,500.00	356.82	
217—General Printing	932.16	750.00	182.16	
218—Office Supplies	682.64	750.00		67.36
219—Addressing and Address Changes	263.72	250.00	13.72	
220—Professional Services	1,624.20	1,000.00	624.20	
221—Bank Charges	350.21	100.00	250.21	
222—Depreciation—Furniture and Fixtures	278.91	250.00	28.91	
223—General Office Expenses	1,131.11	1,000.00	131.11	
Total Headquarter's Expenses	52,806.33	46,250.00	6,638.96	82.63
Less: 30% Charge to Guide	15,841.90	13,875.00	1,966.90	
	36,964.43	32,375.00	4,672.06	82.63
Initiation Fees to Reserve Fund	7,593.45		7,593.45	
Secretary Anniversary Award	237.88		237.88	
Research Fund-Raising Campaign	7,540.60	15,000.00		7,459.40
Improvements and Moving—Cleveland Lab	12,656.59		12,656.59	
Total Expenditures	$169,413.68	$134,150.00	$46,248.65	$10,984.97

Report of Tellers

W. A. Sherbrooke, New York, N. Y., chairman of the Board of Tellers, announced the results of the election of officers for the year 1947 as follows:

BALLOT FOR OFFICERS

	For
President—Baldwin M. Woods	1652
1st Vice President—George L. Tuve	1652
2nd Vice President—A. E. Stacey, Jr	1652
Treasurer—John F. Collins, Jr	1651
Members of Council—M. W. Bishop	1642
Carl F. Boester	1642
Leo Hungerford	1646
R. F. Taylor	1649
E. N. McDonnell	1649

Total Ballots Received	1728
Total Legal Ballots	1652

Invalid Ballots—76. Scattering votes for other candidates.

BALLOT FOR COMMITTEE ON RESEARCH

R. C. Cross	1642
M. K. Fahnestock	1644
John James	1645
F. J. Kurth	1624
T. H. Urdahl	1627

Total Ballots Received	1728
Total Legal Ballots	1652

Invalid Ballots—76. Scattering votes for other candidates.

New Constitution and By-Laws

W. T. Jones, Boston, chairman of the Committee on Constitution and By-Laws, presented proposed revisions to the Constitution of the Society and gave a brief resume of the reasons for the changes which were suggested.

Upon motion of Mr. Jones, duly seconded, it was voted:

THAT the revised Constitution be submitted to the membership by letter ballot for adoption in accordance with *Article C-XVI-Amendments*.

COMMITTEE ON RESEARCH REPORT—1946

By L. P. SAUNDERS, *Chairman*

During 1946 the purchase of the property at 7218 Euclid Ave., Cleveland, Ohio, by the Society provided a permanent Research Laboratory. Almost every feature of the site and the buildings thereon has justified the wisdom of the decision to secure this property, which is ideally suited for research purposes. After several months of strenuous work the research program is once again in full stride with perhaps more work ahead calling for early attention than we have faced for many years.

The Committee on Research fully recognizes and gratefully acknowledges the wholehearted support which the Council has given it in 1946. The enthusiastic response

by the Chapters and the membership at large to the voluntary campaign to reduce the mortgage on the property is encouragement which the Committee appreciates deeply.

Committee Activities: The Chairmen and members of 16 Technical Advisory Committees have shared with the Committee the responsibility of planning and supervising the research program. Fifteen meetings of Technical Advisory Committees have been held between January 1946 and the 53rd Annual Meeting, and 10 Committees are meeting within the next few days. These Committees are the life blood of our organization, and they play a most important part in the research program.

Industry Support: The appointment of C. A. McKeeman, effective February 1, 1946, as assistant to the president, provided liaison between the Society and industry and provided for the solicitation of the funds which are a necessary supplement to general research funds, if we are to carry through any substantial part of the large and varied program before us. We are encouraged by the cordial reception which, he reports, our programs have in general received from the industry and particularly by the financial support afforded to certain specific programs now under way or planned for early attack.

Research Policy: A special Committee appointed by the Research Executive Committee has, under the Chairmanship of E. Holt Gurney, considered to what extent Society research might be expanded and the program broadened. Their report recommended that *Article B-XIII* of the By-Laws, namely, *The work conducted by the Research Laboratory of this Society shall be confined to a determination of the basic or fundamental principles or laws underlying all matters in the science of heating, ventilating and air conditioning,* be retained in its present form with the broad interpretation that fundamental research includes the development of procedures for test codes but under no consideration permits commercial testing.

The Committee on Research suggests that Technical Advisory Committees should be constantly alert in reviewing codes and standards relating to heating, ventilating, and air conditioning in order to submit proposed changes to the Standards Committee of the Council.

Research Regulations: The proposed amendments to the regulations governing the Committee on Research submitted to the membership will, if approved, enable the Committee to extend its cooperative research activities as the occasion demands.

Publication Program: In the Chairman's Annual Report for 1945, Prof. G. L. Tuve drew attention to the urgent need for a change in the Society's publication policy for research reports. We view with satisfaction the recommendations on publication policy made by a special committee and approved by the Council in October. We believe that these changes will greatly increase the interest of most of the membership in the results obtained from Society research.

The two pages devoted to research news which have appeared monthly since August in the JOURNAL SECTION are designed to keep the work of the Committee constantly before the membership.

Future Plans: A minimum budget of $75,000 has been set for 1947 and approved by the Council. If we are able to carry through, with industry support, some of the large programs now under development, this may be increased substantially. Plans are under way to develop a program of research in panel or radiant heating and cooling, to meet the challenge that the Society should assume leadership in solving this important problem.

REPORT OF DIRECTOR OF RESEARCH

By CYRIL TASKER

With the purchase and occupation of the property at 7218 Euclid Ave., Cleveland, Ohio, the hope of some of those who set A.S.H.V.E. Research in motion 26 years ago was fulfilled—the Society owned its Research Laboratory. It is no light task

to move a going laboratory, and the writer wishes to express his thanks to the members of the Laboratory staff for the way in which the work was carried through. We are now in full operation once again, with all essential changes and construction completed. The spirit of the research staff is excellent; every one has been encouraged by the enthusiasm of those who have visited the Laboratory and have seen its possibilities.

The campaign among the Chapters to liquidate the mortgage on the property encourages us to feel that the value of Society Research is being recognized by every member. This is a challenge to us so to serve the membership and industry that we can count on their continued and increased support.

Staff: There were few staff changes in 1946. In July we lost the help of Dr. Allen D. Brandt, who had been assigned to the Laboratory in October 1945, by reason of his resignation from the Public Health Service. An increase of the scientific staff will be needed in the near future, as we activate additional research programs.

Industry Contacts, Liaison and Publicity: The task of establishing and maintaining contact with industry was assigned by the Council in February, 1946, to C. A. McKeeman, who is working in close liaison with the Director of Research and the staff. His visits to the leading industrial organizations and his discussions with officers of the various scientific, trade and industrial associations are beginning to bear fruit in a better understanding of our work, its possibilities and its limitations, and in increased financial participation as new projects are developed and existing projects brought to the attention of the industry.

Talks on the research program have been given before many Chapters of the Society during 1946. Staff members have visited many of the institutions where cooperative research work is or has been in progress.

The excellent facilities available at the Laboratory for meetings of committees have been well utilized since the end of July. The Research Executive Committee met on June 1 in Pittsburgh and on September 21 at the Laboratory. Many Technical Advisory Committees are meeting in Cleveland during the Society's 53rd Annual Meeting.

Some of the highlights of Society's Research have been presented each month since August 1946 in two pages of the Society's JOURNAL.

Our work is receiving greater attention abroad in the technical press of other countries, and our correspondence with foreign countries is increasing. We have had a number of foreign visitors this year and have established liaison with workers abroad by making them corresponding members of some of the Technical Advisory Committees. Senior staff members have continued certain inter-society activities which are of value in our research program.

Guide Activities: Close cooperation has been maintained with the Guide Publication Committee, and all senior staff members have assisted in the review and revision of Guide Chapters.

STUDIES AT THE LABORATORY AND COOPERATING INSTITUTIONS

No report of ordinary length could do more than summarize the activities at the Laboratory and at the cooperating institutions, and those of the Technical Advisory Committees around which Society Research revolves. The details are available in the minutes of the meetings of the various Committees and in the progress reports made by the Laboratory staff and by those responsible for our cooperative studies. The Laboratory staff will be pleased to supply details to Society members and others on request. Those specially interested in certain projects are invited to visit the Laboratory and discuss them with the staff.

Air Flow, Distribution and Friction: Activity in programs classified under this general heading has continued throughout 1946. The Technical Advisory Committee on Air Distribution and Air Friction (Ernest Szekely, chairman) met at the Laboratory

in July to review Laboratory and cooperative research programs. The air friction chart for round ducts, printed in THE GUIDE 1946, has been modified to include smaller ducts and lower volumes, and a paper was published[1] by the Society. Mr. Madison is chairman of the Sub-Committee on Air Friction.

Committee approval was given to the program for study of air friction in round, square and rectangular ducts and in fittings of the type normally used in air distribution systems. As the year closes the erection of the apparatus is nearing completion, and studies will be under way early in 1947. We have to thank several friends in the industry for their valuable assistance in securing many scarce materials and even scarcer equipment for these studies. This investigation should have widespread interest, since the data we hope to secure are fundamental for the design of air distribution systems. Sound and acceptable data should obviate the need for the use of the large factors of safety common in present-day design.

Cooperative Research

1. *Case School of Applied Science (now Case Institute of Technology):* To extend the data already reported from this investigation, which have dealt with the air-stream patterns from various sizes and shapes of straight-flow grilles, nozzles, orifices and slots, further studies have now been made covering directional-flow grilles, perforated panels and very long slots. The static regain in long runs of duct with uniformly distributed outlets has also been investigated.

Wide-angle grilles giving air-stream angles up to 90 deg have been studied ,and equations derived for the throw and the velocity gradients. Perforated panels of 5 to 15 percent free area, in sizes of 1, 2 and 4 sq ft area, as well as complete perforated ceilings have been tested, and methods for determining static regain and the stream angle for such long-run air supply systems have been determined.

A report on the performance of directional-flow grilles, perforated panels, long slots and long duct runs with side outlets is in preparation.

2. *Kansas State College:* Essentially these studies are designed to investigate the downward projection of heated or cooled air streams, and the influence of initial flow conditions and of the surroundings and various structural obstacles upon the behavior of projected streams. The general procedure to date has been to set the initial stream temperature and velocity at desired magnitudes and run temperature and velocity traverses of the downward-projected stream. There have been serious problems in instrumentation especially in the measurement of very low velocities, but the test results to date have been encouraging.

Progress reports have been made covering the results obtained using the outlet of a straight pipe 6½ in. in diameter and 36 in. long as an orifice. The test results have been compared with tentative formulas based on theoretical considerations. Tests have also been made using a converging nozzle with an orifice diameter of 5¾ in. (0.48 ft). A report will be available early in 1947.

Industrial Unit Heater Association: Following several discussions between officers of the *Industrial Unit Heater Association* and its Engineering Committee, members of the Technical Advisory Committee on Air Distribution and members of the Laboratory staff, a detailed program has been presented covering the investigation of the characteristics of air streams produced by downblow unit heaters. The proposal is now awaiting action of the Association.

Throw of Air from Slots and Jets: A paper[2] on the throw of air from slots and jets was published after study by the Technical Advisory Committee. Based on data given

---

[1] Friction Charts for Gases Including Correction for Temperature, Viscosity and Pipe Roughness, by Richard D. Madison and Walter R. Elliot. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*, October 1946, p. 107.)

[2] Throw of Air from Slots and Jets, by Richard D. Madison and Walter R. Elliot. (A.S.H.V.E. JOURNAL SECTION, *Heating Piping & Air Conditioning*, November 1946, p. 108.)

in papers by G. L. Tuve and his colleagues resulting from A.S.H.V.E. cooperative research work at Case Institute, the paper presented some charts designed to simplify problems in air distribution. At the Laboratory H. B. Nottage is reviewing published information on ventilation jets as a preliminary to the development of a research program covering the performance of non-isothermal jets and the general movement of air in ventilated rooms.

Barometric Dampers: A study of so-called *barometric dampers* is being undertaken under the Technical Advisory Committee on Fuels. Since the earlier studies, which were based on the methods which have been used by others to investigate the performance of these appliances, yielded little of real fundamental value, the studies were expanded to cover the examination of the flow of gases in a duct-tee section containing a swinging damper. A report entitled *What About Barometric Dampers,* intended for publication to stimulate interest in these studies, is at present in the hands of the Committee. Another paper entitled *Damper Mechanics* deals with the practical significance of the relation between the position of an automatic damper and the torque which must be applied to balance the damper in each position. It has also been distributed for Committee study.

The continuation of these studies must depend to a large extent on the financial support of those most interested in this type of control.

Industrial (Process) Ventilation: Dr. Brandt, with the assistance of R. J. Steffy and R. G. Huebscher, both of the Laboratory staff, completed two studies during the year. The first, described in a paper[3] presented at the Semi-Annual Meeting of the Society in June, 1946, dealt with energy losses at suction hoods. There were 175 different types of suction openings investigated, the energy losses at the entrance were determined, and coefficients of entry calculated. There was considerable discussion on this paper by those working in the field of industrial ventilation.

The results of the other study are reported[4] in a paper scheduled for presentation at the 53rd Annual Meeting, January 1947. Data were collected on the relationship of the centerline or axial velocity in front of suction openings to the volume rate of air flow on both unflanged and flanged hoods and on hoods having large flat surfaces adjacent to one or more edges. In the main, the tests confirmed Dalla Valle's equation for the centerline flow relationship for unflanged suction openings.

The collection and correlation of information on process ventilation and the collection of codes on the control of occupational diseases were completed prior to Dr. Brandt's resignation from the U. S. Public Health Service. The results of these surveys clearly pointed to the need for establishing factual and dependable data by experimental work. The collection of codes has been loaned to the Michigan Department of Health for analysis by one of its investigators.

Heat Flow and Heat Transfer: Studies under the Technical Advisory Committees on glass, cooling load, and insulation can be classed under this general heading. Substantial progress can be reported during 1946 with three papers published. Several other papers will be available in 1947 as a result of work completed and in progress.

Film Coefficients: A paper[5] on forced convection heat transfer coefficients was presented at the Society's Semi-Annual Meeting in June by two members of the Laboratory's staff. A report covering a continuation of these studies and dealing with laminar and laminar-turbulent boundary layers has been prepared for Committee study. The effect of forced convection on natural convection heat transfer has practical significance, and the Laboratory studies are being continued as time and personnel

---

[3] A.S.H.V.E. RESEARCH REPORT No. 1295—Energy Losses at Suction Hoods, by Allen D. Brandt and Russell J. Steffy. (A.S.H.V.E. TRANSACTIONS, Vol. 52, 1946, p. 205.)

[4] Nature of Air Flow at Suction Openings, by Allen D. Brandt, Russell J. Steffy and Richard G. Huebscher. (See Chapter 1305.)

[5] Forced Convection Heat Transfer Coefficients Along a Flat Surface, by George V. Parmelee and Richard G. Huebscher. (See Chapter No. 1316.)

permit to determine the effect of widely spaced transverse ribs. In the fully equipped draw-through wind tunnel constructed for these studies we have a valuable piece of stand-by equipment; though it will take some time to complete the program, valuable data can be obtained with it in due course.

Solar Radiation Transmission Through Glass: The equipment for the study of solar heat transmission by glass and other materials, designed and constructed by the Laboratory staff, has been completed and test work is under way. Late in the summer we erected, in the rear of the Laboratory grounds and clear of obstructions and shading, a 12 ft by 12 ft test house, 11 ft high with over-hanging roof. It is equipped with electric power and gas, and the special test equipment is mounted on the roof, together with certain instruments for measurement of weather conditions. Inside the test house we have assembled all the control, measuring, and recording instruments needed in these studies. Though the erection was not completed until the late fall, we have been able to collect some data on heat gain from sky radiation through single glass. The winter months will enable us to obtain additional experience in operating the equipment so that a full program can be carried through in the spring and summer of 1947. Among the instruments erected on the roof is an Eppley pyrheliometer for the collection of solar radiation data which may be of value to the U. S. Weather Bureau.

A report on a comparison of experimental and theoretical data on solar heat gain through windows is before the Technical Advisory Committee on Glass for study and possible publication. We have not yet been able to publish much of the information contained in the comprehensive report prepared by Mr. Parmelee late in 1944 but hope to present it in a series of papers during 1947.

Periodic Heat Flow: The mathematical analysis of heat flow through walls and roofs under way at Cornell University under a cooperative research agreement was extended in 1945 to cover composite walls or roofs. The results were given in a paper[6] presented at the Semi-Annual Meeting, 1946. The same authors also presented at this meeting a paper on the sol-air thermometer.[7] The instrument described was designed and constructed to directly determine the temperature in the shade that would be equivalent to the combined effect of air temperature plus sun radiation and sky radiation during sunlit hours, or to the combined effect of air temperature and radiation to the sky at night. The instrument used in the studies at Cornell is now at the Research Laboratory and will be used to collect additional data in connection with the studies on solar radiation transmission discussed above.

At the Laboratory, Mr. Nottage has taken over the correlation of published work on periodic heat flow and the study of methods for the development of tables to give a simple basis for determining the instantaneous rates of heat flow through various solid structures. Such tables would simplify the work of the designing engineer.

Volumetric Specific Heat of Building Materials: The volumetric specific heat is one of the properties of building materials needed in any calculations of periodic heat flow. The Laboratory was therefore assigned the task of collecting and correlating data on this property for the commonly used building materials. A comprehensive report by Mr. Nottage is before the Technical Advisory Committee on Cooling Load. Summary tables will be available shortly for consideration for publication and use in The Guide.

Sub-committees of the Committee on Cooling Load have been working on appliances and lights and on absorptivity. Some of the data obtained will be included in the chapter on cooling load in the Heating, Ventilating, Air Conditioning Guide, 1947, to which many members of the Technical Advisory Committee contributed.

---

[6] A.S.H.V.E. Research Report No. 1299—Periodic Heat Flow—Composite Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. Transactions, Vol. 52, p. 283.)

[7] A.S.H.V.E. Research Report No. 1298—The Sol-Air Thermometer—A New Instrument, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. Transactions, Vol. 52, 1946, p. 271.)

Heat Transfer Coefficients of Evaporating Refrigerants: Studies have continued at Case Institute to determine the effect of the major operating variables upon the refrigerant-side heat transfer coefficient in an evaporator. It has been definitely established in these tests that this surface or film heat transfer coefficient may vary over a range of about 100 to 400, depending chiefly upon the evaporator load, but also upon the length of circuit and the temperature difference between the metal surface and the evaporating refrigerant. If it were possible to design and operate a given evaporator (such as a direct-expansion air cooling coil), so as to insure the maintenance of a refrigerant-side coefficient near the maximum of this wide range, smaller evaporators could be used for a given service. A full understanding of how the very low coefficients may be produced would clear up many operating troubles.

Insulation: A program initiated late in 1945 under the Technical Advisory Committee on Insulation to obtain more accurate and dependable values of the thermal conductivity of building materials has made satisfactory progress. Twelve laboratories, all having equipment conforming to the latest *A.S.T.M.* standards, have agreed to cooperate and have been supplied with samples of corkboard for test. Three laboratories have completed the first tests and submitted the results. All samples, after testing by the laboratory concerned, will be sent to the National Bureau of Standards where check tests will be made. It is expected that the new thermal conductivity values will be ready for THE GUIDE, 1949. The Committee is cooperating with the *A.S.R.E.* Insulation Committee so as to coordinate our test points with the values which the *A.S.R.E.* will publish.

Heat Losses Through Wetted Walls: A paper giving the final results of the studies made at Oregon State College under a cooperative research agreement was presented[8] at the Semi-Annual Meeting in June 1946 by Prof. E. C. Willey.

Environment and Comfort

Comfort Reactions: In 1938 a series[9] of papers was published reporting studies, made in Minneapolis, on the shock experiences, general reactions and summer cooling requirements of 275 workers in an air conditioned office. Since many more data were collected than could be carefully analyzed at the time, arrangements were made with Prof. F. B. Rowley at the University of Minnesota to make a more complete statistical analysis.

His report, under study by the Technical Advisory Committee on Sensations of Comfort, (Prof C. P. Yaglou, *chairman*) has raised a number of questions to which the Committee on Research should find the answers. The results appear to reiterate an important limitation of effective temperature which has troubled investigators in this field for some time. General experience in the past 25 years has indicated that the effective temperature index makes too much allowance for humidity at ordinary temperatures, and not enough allowance in very high temperatures which approach the limit of man's endurance to heat. This, Professor Yaglou suggests, is largely due to the fact that the index takes no account of acclimatization, though in 1922, when the index was developed, it had been assumed that it did.

It has been suggested that the Research Laboratory undertake further experiments to investigate the effective temperature index for humidity and for radiation. In

---

[8] A.S.H.V.E. RESEARCH REPORT No. 1300—Heat Losses Through Wetted Walls, by E. C. Willey, (A.S.H.V.E. TRANSACTIONS, Vol. 52, 1946, p. 297.)

[9] A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337.)

A.S.H.V.E. RESEARCH REPORT No. 1102—Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 571.)

A.S.H.V.E. RESEARCH REPORT No. 1103—General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 591.)

view of the increasing interest in radiant heating systems, there is a moral obligation on the Research Laboratory to establish radiation correction factors.

Shock Effect: Just prior to the outbreak of the war the Society initiated a cooperative research investigation at the University of Illinois Medical School, Chicago, to study the so-called *shock effect.* The tests, which had to be discontinued in 1942, were resumed in the summer of 1946 under the direction of Dr. R. W. Keeton. They have to do with the determination and analysis of the physiological adjustments of human subjects in passing from a hot environment which demands free sweating to one conditioned for comfort, and in passing from a comfortable environment to a hot one.

In the studies made to date five healthy males have been used as subjects. To establish basic data they have been dressed in 90 percent cotton union suits. The results show that the physiological adjustments made by healthy subjects occur very rapidly and place very little strain on the cardiovascular system. Later observations will be made on healthy female subjects and on both male and female subjects wearing typical summer clothing. Some physiologically impaired subjects will be used in the later tests.

These studies have an important bearing on all types of comfort cooling applications including transportation air conditioning. The Society is fortunate in being able to carry on these investigations at the Medical School in Chicago where the University of Illinois has extremely well equipped test rooms and a highly skilled and interested staff.

All Society research investigations dealing with any aspect of physiological reactions are regularly reviewed by the Technical Advisory Committee on Physiological Research of which Dr. C.-E. A. Winslow is chairman.

Air Conditioning in Southern Schools: The Delta Chapter has suggested that the Society undertake the planning and supervision of a series of studies to determine the effect of air conditioning in schools in the south. The proposal is under review by the Committee on Research.

Radiant Heating and Cooling

Interest in radiant heating continues, and the Committee on Research must take action in this field if the Society is to retain its leadership in studies in heating methods and systems. Very little work is being done in this country at present to determine fundamental design data for panel heating systems. Studies have been, or are being made, on specific systems or methods, but the results are limited in their application to the overall problem.

The Technical Advisory Committee on Radiation and Comfort (J. C. Fitts, *chairman*) has defined the scope of its activities as the *consideration of problems in connection with radiation as it affects comfort and the development of heat transfer design data as it affects these problems.* Radiation was defined as *the net exchange of radiant energy between an individual and his surroundings.* The Committee indicated that the problems were not restricted to any type or types of heating or cooling systems. The Laboratory has continued its survey of technical literature and has canvassed college and university laboratories throughout the country to determine whether research work in radiant heating or cooling was in progress or planned. In the main the results were negative except for one or two institutions where investigations are under way with or without the help of the Society.

Field Studies: The Laboratory has prepared the ground for conducting field tests of existing radiant heating systems in the Cleveland area in the hope of developing data for immediate use. A special surface thermocouple has been constructed to determine wall surface temperatures and an aspirated thermocouple made to eliminate the radiation effect upon air temperature determinations. A number of suitable homes have been located by C. M. Humphreys and plans made to initiate tests.

Kansas State College: These studies are designed to investigate the relationship between radiant heating and cooling, and human comfort and well-being. A test cubicle, 10½ ft sq, has been built within a larger room and equipped to enable the floor and portions of the walls and ceiling to be heated or cooled at will. Conditioned air can be supplied as desired to the test room and to the space between the test room and the enclosing structure. Details of the construction and instrumentation are given in progress reports made by Prof. Linn Helander, under whose direction the studies are proceeding.

It is intended to correlate thresholds of comfort with mean radiant temperatures for various distributions of panel surfaces and varying air temperatures at a constant velocity of about 15 fpm. Skin temperatures will be measured, together with the temperatures of surrounding surfaces and the mean radiant temperature. A special radiometer has also been constructed. The judgment of an individual's comfort sensations will be accompanied by a determination of the physical quantities which represent the influence of the surroundings upon the energy balance for his body.

A progress report covering an analytical study of the emissivity of unheated surfaces as a factor in the design of radiant heating systems has been submitted to the Technical Advisory Committee for study.

Cooperation With Other Organizations: Acting for the chairman of the Committee on Research, the director of research and Mr. McKeeman met with the Research Committee of I=B=R in Chicago on December 10, 1946, to discuss the possibility of cooperation in the field of panel heating research. It was finally proposed that representatives of all groups having an interest in panel heating or cooling be invited to a meeting in Cleveland early in March, 1947. This meeting would, it was hoped, crystallize the various ideas on research into panel heating. The Laboratory would present a suggested research program for the development of basic data needed in the design, installation and operation of radiant heating systems with estimates of the probable length of time involved and some approximation of cost. It was agreed that most of the industries referred to as being interested in panel research had representative trade organizations which should be asked to delegate official representatives to attend the meeting. The Laboratory was instructed to proceed with the preliminary plans for this meeting and for the presentation of the program.

Miscellaneous Studies

Air Cleaning: In January, 1946, the Technical Advisory Committee on Air Cleaning (R. S. Dill, chairman) voted that a program of research and test work be developed with the ultimate aim of recommending sound and acceptable codes for the testing and rating of air cleaning devices. It was suggested that funds be solicited from both manufacturers and users to enable a qualified investigator to be engaged for full-time work at the Laboratory A tentative program was drawn up and a proposal circulated to those likely to be interested. At the year's end sufficient funds were in hand or pledged to enable a start to be made on this program. These studies are of considerable importance to the future of air conditioning, and the problem is receiving considerable attention at the present time not only in America but also abroad. There is apparently considerable dissatisfaction with the present status of the testing and rating of air cleaning devices.

Psychrometry: The studies on the measurement of the dew-point of air-water vapor mixtures, under way at the University of Toronto under a cooperative research agreement, have reached the stage where a completely automatic dew-point hygrometer has been designed and built and its operation tested over a wide range of conditions. The report on this work states that "the most noteworthy conclusion drawn from observations with the instrument is that an apparently large difference may be observed between the theoretical and actual dew-point of moist air." Further

work would seem to be advisable and the University of Toronto is planning to continue these studies.

In October the Laboratory presented a paper on psychrometric instrumentation.[10] Its purpose was to provoke discussion and study of this important branch of air conditioning.

A sub-committee of the International Joint Committee on Psychrometric Data has been examining the differences on certain fundamental concepts and constants which apparently exist among the organizations and countries represented on the Joint Committee, as a preliminary to the construction of a new set of international psychrometric tables.

Weather Design Conditions: The Committee on Weather Design Conditions (T. H. Urdahl, chairman), has made a complete analysis of the annual weather data for Detroit, Mich., over the five-year period 1935-1939, inclusive. The major part of the credit for this analysis should go to J. C. Albright,[11] a member of the Committee. The report is being carefully studied so that a method of presentation acceptable to those interested in such data can be set up as a model for the other 116 stations for which complete data are available. The importance of accurate weather data and of the incidence and co-incidence of temperatures, relative humidities, and wind velocities is being more fully realized day by day. With the data available the Society can make a distinct contribution to a better appreciation of the significance of design conditions.

Corrosion: The activities of the Technical Advisory Committee on Corrosion (Leo F. Collins, chairman) have been devoted exclusively to the preparation of a proposed chapter on water-formed deposits and corrosion for THE GUIDE. It covers problems of scale, sludge and slime formation in all types of heating and ventilating equipment employing water, and discusses corrosion on the interior and exterior surfaces of such equipment.

Sorbents: Much of the activity of the Technical Advisory Committee on Sorbents (John Everetts, Jr., chairman), has consisted of the revision and rewriting of Chapter 38, Dehumidification by Sorbent Materials of THE GUIDE 1947.

Sound Control: Following a meeting of the Technical Advisory Committee (R. D. Madison, chairman) in July, arrangements were made for Committee participation in studies to be made by the U. S. Navy to determine the best methods of acoustical testing of ventilating fans and systems and to establish a method of predicting the acoustical performance of ventilating systems. A research program, drawn up by the Bureau of Ships, was submitted for Committee study and comment, and work is now under way at the Material Laboratory, U. S. Naval Shipyard, New York.

Through Committee action, the help of the Bureau of Standards and *American Standards Association* was also sought in the problem of the testing and calibrating of sound measuring equipment to improve correlation between tests made with various instruments and at differing institutions.

ACKNOWLEDGMENT

The writer respectfully acknowledges both on behalf of the Research Laboratory staff, and personally, the encouragement and help given during the year by the Chairmen and members of the Technical Advisory Committees, by members of the Committee on Research and especially by the Chairman and the members of the Research Executive Committee. They have made a difficult and yet, we believe, an important year more effective than it might have been.

---

[10] Psychrometric Instrumentation (A review of the accuracy of psychrometric instruments and their application.) (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning,* October 1946, p. 101.)

[11] Analysis of Summer Weather Data in the United States, by J. C. Albright. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 397.)

STATEMENT OF INCOME AND EXPENSES

(For the Fiscal Year Ended October 31, 1946)

INCOME

FROM A.S.H.V.E.				
40 per cent of 1946 and prior year's dues collected				$29,958.83
CONTRIBUTIONS—PER SCHEDULE				
General			$ 8,282.00	
Earmarked				
Contributions received	$14,950.00			
Add: Deferred from prior year	2,874.62	$17,824.62		
Less: Deferred to future operations		7,146.15	10,678.47	18,960.47
INTEREST FROM BANK DEPOSITS				3.19
TOTAL INCOME				$48,922.49

EXPENSES

CHAIRMAN AND COMMITTEE				
Traveling		$ 653.54		
Chairman's office		15.25		
Professional services		250.00	$ 918.79	
RESEARCH LABORATORY				
Salaries—Administrative and Technical		$32,791.41		
Clerical		6,459.47		
Traveling		1,833.45		
Postage		394.51		
Telephone and Telegraph		815.75		
Office Expense and Supplies		661.13		
Insurance		67.94		
Printing and Mimeographing		409.68		
Library and Periodicals		52.05		
Laboratory Materials and Supplies		2,857.94		
Depreciation—Equipment and Fixtures		1,319.16		
Unallocated		864.50		
		$48,526.99		
BUILDING OPERATIONS				
Cleveland Rent	$ 1,053.41			
Real Estate Taxes	1,160.79			
Depreciation—Building	1,036.66			
Interest on Mortgage	947.19			
Property Insurance	371.93			
Janitor	748.66			
Heating	539.61			
Electricity	464.65			
Building Maintenance	196.42			
Water	84.60			
Gas	79.42	6,683.34	55,210.33	
COOPERATIVE RESEARCH				
Case School of Applied Science		$ 1,000.00		
Kansas State College		2,200.00		
Oregon State College		100.00		
State University of Iowa		250.00		
Texas A & M College		250.00		
University of Toronto		773.00	4,573.00	60,702.12
EXCESS OF EXPENSES OVER INCOME				$11,779.63

PARTICIPATION IN 1946 RESEARCH PROGRAM

*Air Devices, Inc.
Airtemp, Div. of Chrysler Corp.
*Aluminum Venetian Blind Co.
*American Air Filter Co., Inc.
*American Blower Corp.
American Rolling Mill Co., The
*Anemostat Corp. of America
*April Showers Co.
Barnes & Jones, Inc.
*Bayley Blower Co.
*Bell Telephone Laboratories
Bethlehem Steel Co.
*Blue Ridge Glass Corp.
Brunner Mfg. Co.
*Buffalo Forge Co.
*A. M. Byers Co., The
Carrier Corp.
Chamberlin Co. of America
Chicago Pump Co.
Clarage Fan Co.
Crane Co.
*Detroit Edison Co., The
W. H. Driscoll
Duquesne Light Co.
Delmar C. Evans
*Farr Co.
*Ford Motor Co.
Forslund Pump & Machinery Co.
Frick Company (Inc.)
*Frigidaire Div., General Motors Corp.
*General Motors Corp.
G & O Mfg. Co., The
Hays Corp.
**Heating, Piping and Air Conditioning Contractors National Association*
Hoffman Specialty Co.
*Holcomb & Hoke Mfg. Co.
Ilg Electric Ventilating Co.
*Illinois Engineering Co.
Jenkins Bros.
*Johns-Manville
Johnson Service Co.
Keeney Publishing Co.
Kewanee Boiler Corp.
Kinetic Chemicals, Inc.
Kramer Trenton Co.
*La-Del Div., Joy Mfg. Co.
*Libbey-Owens-Ford Glass Co.
R. C. Mahon Co., The
Jas. P. Marsh Corp.
McDonnell & Miller, Inc.
McQuay, Inc.
Mellish & Murray Co.
*Minneapolis-Honeywell Regulator Co.
Modine Mfg. Co.
*Narowetz Heating & Ventilating Co.
Nash Engineering Co., The
National Radiator Co., The
*National Tube Co.
John J. Nesbitt, Inc.
*Perfex Corp.
Pipe Fabrication Institute
*Pittsburgh Corning Corp.
*Pittsburgh Plate Glass Co.
*Pyle-National Co., The
Raisler Corp.
W. R. Rhoton Co.
*F. C. Russell Co., The
*Sarco Co., Inc.
*Servel, Inc.
Spencer Thermostat Co.
Surface Combustion Corp.
Taylor Instrument Cos.
Timken Silent-Automatic Div., The Timken-Detroit Axle Co.
*Trade-Wind Motorfans, Inc.
Trane Co., The
United States Steel Corp. of Delaware (Carnegie-Illinois Steel Corp.)
*United States Testing Co., Inc.
Webster Engineering Co., The
W. H. Wheeler, Inc.
*L. J. Wing Mfg. Co.
York Corp.
*Young Radiator Co.

*Earmarked Contributions.

INSTITUTIONS COOPERATING WITH THE COMMITTEE ON RESEARCH

Agricultural and Mechanical College of Texas, College Station, Tex.: The Effect of Secondary Turbulence on the Friction in a Flowing Stream of Water. **Case School of Applied Science,** (now Case Institute of Technology), Cleveland, Ohio: Air Distribution in Rooms; Heat Transfer Coefficients of Freon Refrigerants. **Cornell University,** Ithaca, N. Y.: Periodic Heat Flow through Composite Walls or Roofs. **Kansas State College,** Manhattan, Kans.: Projection of Heated and Cooled Air Streams; Experimental Studies of Radiant Heating and Cooling. **Oregon State College,** Corvallis, Ore.: Heat Transfer through Wetted Walls. **University of California,** Berkeley, Calif.: Cooling Tower Design and Performance. **University of Illinois**—College of Medicine, Chicago, Ill.: Physiological Adjustments of Human Beings to Rapid Changes in Environment. **University of**

Minnesota, Minneapolis, Minn.: Statistical Analysis of Variables Entering into Sensory Reactions to Summer Air Conditioning. **University of Toronto,** Toronto, Ontario, Canada.: Moisture Condensation on a Surface and the Measurement of the Dewpoint.

RESEARCH PAPERS—1946

1. The Sol-Air Thermometer—A New Instrument, by C. O. Mackey and L. T. Wright, Jr. (Ithaca, N. Y.) (A. S. H. V. E. TRANSACTIONS, Vol. 52, 1946, P. 271).
2. Forced Convection Heat Transfer from Flat Surfaces, by George V. Parmelee and Richard G. Huebscher (Cleveland) (see Chapter No. 1316).
3. Heat Losses Through Wetted Walls, by E. C. Willey (Corvallis, Oregon) (A. S. H. V. E. TRANSACTIONS, Vol. 52, 1946, P. 297).
4. Periodic Heat Flow—Composite Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (Ithaca, N. Y.) (A. S. H. V. E. TRANSACTIONS, Vol. 52, 1946, P. 283).
5. Energy Losses at Suction Hoods, by Allen D. Brandt and Russell J. Steffy (Cleveland) (A. S. H. V. E. TRANSACTIONS, Vol. 52, 1946, P. 205).
6. Psychrometric Instrumentation, by A. S. H. V. E. Research Laboratory (Cleveland) (October 1946, JOURNAL SECTION).
7. Nature of Air Flow at Suction Openings, by Allen D. Brandt, Russell J. Steffy and Richard G. Huebscher (Cleveland) (see Chapter No. 1305).

SECOND SESSION—TUESDAY, JANUARY 28, 9:45 A.M.

The second technical session convened at 9:45 a.m. January 28, in the Ballroom of Hotel Statler, Cleveland, Ohio, with Second Vice Pres. G. L. Tuve, Cleveland, presiding.

Chairman Tuve stated that the first order of business was to consider the proposed amendments to the Regulations Governing the Committee on Research which were endorsed by the Council and submitted to members of the Society in accordance with **Article B-I**, *Section 10* of the By-Laws as follows:

Article V—Government

Section 6. Headings of Papers—All papers, findings, or reports resulting from the work of the Committee on Research shall, when published by the Society, be headed as follows:

(*a*) If the paper, finding, or report is the result of work at the Society's Research Laboratory, the following statement shall be used: "This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland, Ohio."

(*b*) If the paper, finding, or report is the result of cooperative work with some institution or laboratory other than the Society's Research Laboratory, the following statement shall be used: "This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with (name of institution)."

Article VI—Patents

Section 1. If any invention or discovery is made by a member or members of the research staff of the Society which is deemed worthy of patent application by the Committee on Research, such application shall be made in the name of the inventor and assigned to the Society. When issued the patent shall be held by the Society in trust for the benefit of the public.

Section 2. If any patentable invention or discovery is made under cooperative research agreements between the Society and a University or College, the Committee

on Research shall determine the conditions under which patents shall be controlled so as to produce the greatest benefit to the Society and the public.

Secretary A. V. Hutchinson explained that the changes in **Article V—Government** were made necessary by the transfer of the Research Laboratory from Pittsburgh to Cleveland. He stated that the amendments to **Article VI—Patents** were also made necessary by the transfer of the Laboratory from Pittsburgh, where it was necessary to conform to certain rules of the Department of the Interior which provided the laboratory space.

Chairman Tuve explained that the change in **Article VI—Patents** *Section 2,* made it easier to meet the requirements of various institutions conducting research in cooperation with the Society.

On motion of H. M. Hart, Chicago, Ill., seconded by R. C. Thomas, Norfolk, Va., it was

> VOTED: That the proposed amendments to the Regulation Governing the Committee on Research be adopted.

Chairman Tuve called on T. F. Rockwell, Pittsburgh, Pa., Chairman of the Guide Publication Committee who presented the report of his Committee which described the procedure of compiling the HEATING VENTILATING AIR CONDITIONING GUIDE 1947 and referred to the principal changes which were described in the preface.

He said that the Committee offered the following comments and recommendations:

> The practice of having one-third of the Guide Publication Committee appointed each year had proved to be beneficial in preserving continuity of policy. This continuity makes it possible to plan for extended revisions which may require more time than is obtainable in any one year. Typical of this long range planning was the appointment during the past year of a committee which was requested to prepare a new chapter on Automatic Controls for submission to the next Guide Publication Committee. With the director of research serving as an ex-officio member of the Committee, information regarding the latest research work of the Society could be incorporated more readily.
>
> It is recognized that occasionally information, which is of value to certain groups or individuals, is not continued in each edition of THE GUIDE because it may lack general interest. It is, therefore, suggested that an index might be prepared to indicate the last edition of THE GUIDE in which such information appeared.
>
> In the preparation of this reference book for designing heating and ventilating systems the Committee is always faced with the problem of trying to satisfy those who wish to have an over-simplified presentation which can be understood by the greatest number of readers and those who desire a more rigorous and scientific treatment of material included in THE GUIDE.

The Chairman thanked the Guide Publication Committee for the work done in preparing THE GUIDE 1947 and also for the suggestions for future committees which were contained in the report. He recommended that each member of the Society should consider himself a potential contributor to THE GUIDE.

Three papers were presented and Chairman Tuve thanked the authors for their presentations. The session was adjourned at 12:00 noon.

THIRD SESSION—TUESDAY, JANUARY 28, 2:00 P.M.

With the Ballroom of Hotel Statler jammed to the rafters, Prof. C. F. Kayan, New York, N. Y., presided as chairman of the forum on panel and radiant heating.

Panel Heating Symposium

Professor Kayan referred to panel and radiant heating as a subject which seemed to have an interest comparable with that taken in the heat pump, the gas turbine, or rocket flight. He stated that the object of the present session was to bring out useful information and to clarify Society thinking on the subject of panel and radiant heating. Professor Kayan announced that for orderly consideration of the subject it had been planned that five divisions of the subject be presented by five Society members, H. H. Angus, C. F. Boester, C.-E. A. Winslow, W. Bruce Morrison, and R. L. Byers, and that following this presentation an opportunity be offered the audience to ask questions and to participate in the discussion.

Is There a Difference Between Panel Heating and Radiant Heating? H. H. ANGUS, Toronto, Can.

We can best approach this subject by reviewing general data so as to bring out the salient points. The figures to be given are approximate only as many factors enter into them, but they are close enough for purposes of this discussion which refers to problems in our industry.

The normal rate of heat production in an average sized sedentary individual is around 400 Btuh. Of this 300 to 320 Btuh are given off by radiation and convection from the external body surface. In still air the radiation loss will be about 190 Btuh and the convection loss about 120 Btuh. If we have high velocity of air the loss by convection will be greatly increased and may go as high as 250 Btuh in which case comfort may be obtained with a radiation loss of 50 Btu.

The area of the body of an average individual is about 19½ sq ft for convection and 15½ sq ft for radiation as some sections of the body such as the legs radiate their heat to other portions.

The rate of heat loss by convection depends on the average temperature difference between the surface of the body and the surrounding air, also on the rate of air motion.

The rate of heat loss by radiation depends on the exposed surface of the body and upon the difference between the mean surface temperature of the body and the mean surface temperature of the surrounding walls, called the *mean radiant temperature* and abbreviated MRT.

The required rate of heat loss can result from a relatively low air temperature and a relatively high MRT as illustrated by a person standing in the sunshine on a fairly cold day. On the other hand the rate of heat loss can result from a relatively high air temperature and a relatively low MRT if, for instance, a person is in a warm room in line with the rays from a radiating surface.

In a conventional radiator type heating system the exposed radiators emit about half of their heat by radiation and half by convection. Convection units emit almost all of their heat by convection.

We should also remember that the intensity of heat radiation varies inversely as the square of the distance from the source of radiation.

The heat received from the rays of the sun is generally referred to as solar heating by engineers and will not be considered here.

In industry there are many examples of heating by radiation from high temperature surfaces, a drying oven being one example.

In open air schools and similar buildings the occupant is surrounded by outside air at relatively low temperature and is exposed to the heat rays from a high temperature surface. Such systems are in general use in many parts of Europe and are definite examples of radiant heating.

Our discussion today is to be limited to closed rooms in which the occupant is surrounded by walls, floor, and ceiling.

In conventional systems, heating is obtained by radiation and partly by convection. As we go a step farther and get more effect by radiation and less by convection we may consider whether a distinctive name should be given to the system.

In mild climates comfort is obtained if the occupant can be exposed to direct rays from a source of heat. In colder climates it is necessary to heat the walls or surrounding air to some extent. This can be illustrated by a person standing in the sun. Down to a certain temperature in still air he will be quite comfortable but below this temperature he will not be comfortable.

Our question then is whether we should refer to heating systems under two names or whether we should have one name to cover all heating systems of this type for systems used in our industry. A starting point may be the definitions given in THE GUIDE in which we note a distinction in the definitions as follows:

Panel Heating: A heating system in which heat is transmitted by both radiation and convection from panel surfaces to both air and surrounding surfaces.

Radiant Heating: A heating system in which only the heat radiated from panels is effective in providing the heating requirements. The term *radiant heating* is frequently used to include both panel and radiant heating.

What Heating Mediums for Panel and Radiant Heating Systems? C. F. BOESTER, Lafayette, Ind.

My remarks are predicated on the assumption that design conditions have been established requiring physical control of one or more surface temperatures of an enclosed space. My remarks are also qualified that such controls are accomplished behind the surface or surfaces involved because you also can affect and control surface temperature by other more conventional means.

The popular mediums for panel heating at the present time are: heated air circulated through the spaces behind the surfaces involved; and heated water circulated through pipe coils in the walls, ceilings or floor surfaces.

There is also the means of controlling the surface temperature electrically by the use, for example, of a rubber film such as was seen on display at the Exposition. This is a compound material which has electrical resistance but a film coated on glass may be used. Likewise, it is possible to use a foil tape having resistance characteristics and place it behind the surfaces involved. Resistance heating, as far as the panel surfaces are concerned, is not too popular except in areas having favorable power rates.

Economically, as one experiment shows, the room enclosure should be low in thermal capacity. Then by placing the heating element in four or possibly all six surfaces, the enclosure may be brought instantaneously to design conditions by the control switch, after which the air thermostat becomes effective as air temperature builds up. It is possible to reduce the energy requirements materially by using heat only in the space being used.

There are several applications of the heat pump in which the hot refrigerant gas from the heat pump cycle is discharged to coils comprising the heat exchange surface.

Panel heating may also be accomplished by use of resistances located in the stud space. Air from the resistances in such a system moves upward within the wall and enters the room at the ceiling line and returns at the baseboard. It is also

feasible to divide the stud space and have the warm air rise in one passage and return in the other.

There are several installations employing Perkins' tubes which are pipes placed vertically in the stud space and sealed at the end. A vacuum is drawn on the system and steam rises in the tubes. They are operated at steam temperature.

There are a number of installations in which pipe coils maintained at steam temperatures by hot water are placed in the stud space. Intelligent use of insulation can materially affect the control of surface temperatures, and it would be recommended that perhaps two kinds of insulation be employed.

There are many problems involved in the mechanical installation of panel systems. Limitation of plaster drying is one of them. My personal experience is quite limited but I have had the opportunity of wide observation of work of others—the organization with which I am affiliated spending some $60,000 in the study of panel heating. Strictly as a personal opinion, I believe that we can get the greatest performance out of a ceiling installation; roughly one square foot of ceiling surface will do the work of two of the floor.

From the mechanical viewpoint and cost of the installation, the simplest installation would be the ceiling type. Small diameter tubing would be preferable on three or two and a half inch centers, depending on the design conditions. Proper headers should be developed to minimize pressure drop. A low volume of water would reduce the hazard of freezing. As a matter of fact small diameter tubing in some experiments was not damaged by freezing. No expansion tank is necessary on such a small system.

One advantage from a control viewpoint gained by use of small diameter tubing is that the fly-wheel effect is minimized and this aids in obtaining control. Also there will be lower first cost for the materials involved and lower installation labor costs.

What Is the Comfort Temperature in Panel-Heated Rooms? C.-E. A. WINSLOW, New Haven, Conn.

The question I have been asked to discuss is not susceptible of any easy or uniform answer.

Comfort—under moderate or cool temperature conditions—is determined by air temperature and mean radiant temperature; and, with low air movement, the two factors are roughly of equal importance.

Mr. Angus has quoted you the figures from THE GUIDE showing that we ordinarily lose more heat by radiation than by convection. And that is true in an ordinary room which is convection heated and in which all the surfaces are at lower temperature than the air. It would, for instance, most emphatically not be true in this room where the wall surfaces are about as warm as the air and where, as far as you in the audience are concerned, radiation in the lateral direction is entirely cut off. You cannot exchange radiation with each other because your bodies are all of the same surface temperature and you can not get any cooler except up or down, so that in a hall of this kind convection is overwhelmingly important. But it is always true that the operative temperature, the thing that really determines comfort is established very closely by the mean between mean radiant and air temperature, provided that the air movement is low. Of course, if air movement is high, the convective heat loss becomes greater while the radiant heat interchange is unaffected. Therefore, the more air movement you have, the greater is the relative importance of air temperature.

In a panel-heated room, however, air movement should be minimal. If we assume that it is 20 fpm, or less (what we commonly call still air) we may take the operative temperature, which determines comfort, as approximately the mean of air temperature and mean radiant temperature. The formulas governing the relations between air

temperature and mean radiant temperature have been worked out graphically by Bedford and Warner.[12]

The subject is complicated, of course, by the fact that, in practice, the mean radiant temperature of a panel-heated room is determined not only by the high temperature of the panelled walls but also by the low temperature of one or more cold exterior walls. What actually happens may be illustrated by these simple examples.

Let us assume a cubical room, with panels in floor and ceiling and with one outside wall whose mean surface temperature, including window surfaces is 55 F, which might occur with two good sized windows in cold weather. Assume also that the mean temperature of ceiling and floor surfaces is 75 F. The mean radiant temperature will then be 55 F (for outside wall) plus 2×75 F for the floor and ceiling plus 3×68 F for the three interior walls, divided by 6 or 68.2 F. To produce an operative temperature of 68 F, the air temperature should be 67.8 F.

If we assume a mean temperature of 80 F for the floor and ceiling, with outside conditions the same, the mean radiant temperature would be 55 F plus 2×80 F plus 3×68 F divided by 6 or 69.8 F; and the air temperature should be 66.2 F.

If the mean floor and ceiling temperature were 85 F the mean radiant temperature would be 71.5 F and the air temperature could be lowered to 64.5 F.

To take a more extreme condition—such as might occur in a corner classroom of a school with large window areas in cold weather—let us assume that two outside walls have a mean temperature of 55 F; and that the mean temperature of floor and ceiling is 85 F. Then the mean radiant temperature would be 2×55 F plus 2×68 F plus 2×85 F, or 69.3 F. The corresponding air temperature for comfort would be 66.7 F.

The point of practical importance is that the essential function of panel heating is to balance through warm walls the heat loss from cold walls. Under practical conditions, the spread between air temperature and mean radiant temperature cannot be very great, since the only way in which a high differential can be maintained is by artificially cooling air which we do in our experimental laboratories but which would be manifestly absurd in practice. Therefore, we cannot assume any practical condition in which air temperature will be more than 2-3 deg below the normal comfort temperature of 68 F.

Where radiant heating is not employed, however, it is clear that the air temperature must be above 68 F in order to balance cold walls and windows. With convection heating alone, the air and the two interior walls, and the floor and ceiling, under the last of the three conditions cited, would have to be 70.6 F to give an operative temperature of 68 F (2×55 F+10×70.6 F÷12).

Finally we must bear in mind that the effect of radiant temperature depends on the angle of exposure of an individual to a given surface. The person sitting near a window will be much colder than a person sitting near a relatively warm wall. Differential distribution of radiant heat must be worked out to meet this situation; just as in convection heating the warm air may be supplied in approximation to the cold wall.

Does a Panel or Radiant Heating System Save Fuel? W. BRUCE MORRISON, Portland, Ore.

In order to make the proper comparison between heating systems, all variable conditions in the buildings should be eliminated. An exaggerated example showing the effect of different conditions is obtained by comparison of two buildings both of 10,000 sq ft floor area, one building of a 100 ft x 100 ft size and one of a 20 ft x 500 ft size. A floor panel heating system in the square building of 100 ft x 100 ft size would no doubt require less fuel than a conventional heating system; but a floor

[12] Bedford and Warner (nomograph), *Journal of Hygiene*, 34:458-473, 1934.

panel heating system in the long building of a 20 ft x 500 ft size might require more fuel than a conventional system. The reason for this is that the larger perimeter losses from the floor in the long building would require a disproportionate amount of fuel.

For the purpose of simplifying this introductory talk I will discuss first an average residence or small building with an area of approximately 1500 sq ft having low ceilings. Assuming a floor panel heating system in a small building of this type, we find that the increased ground losses add possibly 10 percent to the computed heat loss unless extraordinary precautions are used. To offset this increased heat loss, we have the fuel saving which may be realized by the reduction of temperature within the building. The saving which may result from this reduction in temperature approximates the increased floor loss thereby resulting in very little advantage for the panel heating system. Let us now examine a ceiling panel or wall panel system with panels on exposed walls. We find that the increased heat losses from panel areas approximate 6 percent of the total calculated heat loss. To offset the increased loss we, of course, have the reduction in temperature which might provide a saving of approximately 10 percent. These figures indicate to me that there is not too much to be gained in fuel heating costs by the use of panel heating in an average home.

For commercial buildings I am basing my comparison on buildings with 10,000 sq ft or more of floor area with ceilings 24 ft or higher and the use of a floor panel heating system as compared to a conventional unit heater system. With a floor panel heating system the stratification of heat at the higher levels with its attendant increased heat loss is avoided. Also equal comfort conditions can usually be maintained with lower air temperature by using a panel system. An analysis would indicate that sizeable reductions in fuel costs may be realized in larger buildings using floor panel heating and maintaining comfort conditions equal to those with a conventional unit heater system.

In conclusion I wish to state that my observations have indicated that fuel savings to be realized by installing panel heating in residences or small commercial buildings are relatively minor; but that the installation of panel heating in large commercial buildings may provide substantial savings in fuel.

Control Requirements for Panel or Radiant Heating Systems: R. L. BYERS, Cleveland, Ohio.

The temperature control for radiant heating might be defined as the control of the heat flow into a building to maintain comfort.

Several factors, inherent in radiant heating systems, need consideration: (1) The time lag of the heating system or the time until the panel surface will respond to a desired change in Btu output; (2) The time lag of the structure or time until the inside temperature of a building will reflect a change in outside temperature; (3) Effect of solar heat; (4) Effect of wind.

The heat supplied to a building may be controlled by a room thermostat to maintain a certain inside temperature or by a device that will proportion the amount of heat supplied in relation to the outside temperature. A room thermostat may be used in conjunction with an outside controller to stop or reduce the flow of heat in the system, or to a zone or a room if overheating occurs. Solar heat, wind or internal heat gain may make this necessary.

These control systems may be used to modulate the temperature of the heating medium which is supplied continuously or they may be used to control the flow of a constant temperature heating medium. It seems necessary, therefore, that we must analyze a building and its heating system, to determine the proper type of controls.

Let us now consider a few hypothetical cases:

A building constructed of sheet steel is to be heated by a heavy concrete floor slab. The building will respond very rapidly to a change in outside temperature or solar heat but the floor slab will

respond very slowly and a wide fluctuation of inside temperature will result. This combination of short time lag of structure and a long time lag in heating system should be avoided if a good comfort condition is desired.

Consider a masonry building to be heated by a concrete floor. Here we have a structure and a system both having approximately the same time lag and both having a long time lag. An outside controller is indicated because the controller will sense the change in outside temperature and increase the heat flow. The flow will then respond in sufficient time to meet the requirements of the structure.

Consider a masonry building to be heated by a lath and plaster heating panel, a long-time lag on the structure and a short-time lag in the panels. An inside thermostat may be used because the panel response is much more rapid than the structure and will produce heat in time to prevent a wide fluctuation in room temperature. An outside controller on the other hand may cause a supply of heat to the room too soon and thereby cause over-heating.

We may, therefore, draw the following conclusions:

1. The time lag of the heating system should not be greater than the time lag of the structure.

2. If the time lag of the structure and time lag of the system are the same, an outside control might be indicated.

3. If the time lag of the system is short and that of the building is long, an inside thermostat may be used satisfactorily.

Let us not permit our theory, however, to lead us into design of too complicated control systems. Systems must be kept simple in regard to installation, operation, and adjustment. In small systems controls must be simple so that an average layman may understand them and be able to make necessary adjustments. We do not want to use formulas, slide rules, or tables to adjust the equipment or find it necessary to call a man out of bed 15 miles away from the installation on a zero night to come and make an adjustment.

The cost of controls must bear some relationship to the cost of the system or the quality of the job.

If, for instance, a five-room house is to sell for $10,000 in Cleveland, complete with lot, utilities, etc., the panel heating system cannot cost over $850. How can we install a temperature control system to cost $200 out of $850? Unless we can obtain control by means of a room thermostat and a limit switch, it will be impossible to build a house with a panel heating system.

Following the period of analyzing we have been going through, most of us are going to learn best by practical application and time will be our teacher.

DISCUSSION

F. E. Giesecke, New Braunfels, Tex. (Written): Panel Heating had its origin in England about 1907, when Prof. A. H. Barker noticed that rooms whose walls contained flues leading up from fireplaces on the lower stories were much more comfortable than rooms whose walls did not contain such flues. Following this experience he conceived the idea of artificially heating the walls of a room so that their surfaces would radiate heat to the room. He was granted a patent on this new method of heating which he named *Panel Warming*, but as he was not commercially inclined he sold his patent to the firm of R. Crittall & Co., Ltd. This firm developed the discovery. The system was known first as the Crittall system of heating, and later as panel warming, or panel heating, or radiant heating.

Panel heating or radiant heating made fairly rapid progress in England, Switzerland, Germany, and other European countries, but progress in the United States was quite slow, partly because air conditioning was being developed by Mr. Carrier at that time and partly because a royalty was being demanded on the installation of the Crittall system of heating. However, during recent years much publicity has been given to panel heating or radiant heating in the United States and in Canada

and it is possible that some persons have been led to believe that there is a fundamental difference between panel heating or radiant heating and the older methods of heating.

There is no fundamental difference between panel heating and radiator type of heating; in both systems the heat is delivered to the room partly by radiation and partly by air convection currents. The difference between the two systems is that in panel heating a larger portion of the total heat delivered to the room is delivered by radiation than is the case in radiator heating.

The difference between panel heating and warm air heating is less than is generally assumed. In a warm-air heating system, heat is delivered to the room by air convection currents. The warm air rises to the ceiling and floats along the ceiling. If the ceiling is well insulated, the under surface of the ceiling will be only a few degrees cooler than the warm air in contact with it; the ceiling will then become the heating panel for the room and the system may be designated as a warm-air panel-heating system.

The principal contribution which the development of panel-heating has made to the science and art of heating and ventilating is that we now recognize and appreciate the important part which radiant energy has in the heating and cooling of buildings to a much greater extent than we did before. We realize now that every point of every surface radiates energy in all directions and that every room is completely filled with radiant energy moving in all directions with very high velocities and that we are continually immersed in and surrounded by radiant energy.

We realize also that panel heating systems do not heat the occupants of a room and that panel cooling systems do not cool them. Panel heating systems and panel cooling systems, when properly operated, produce satisfactory thermal conditions within the room; *i. e.*, these systems produce the mean temperature of the air within the room and the mean temperature of the surfaces enclosing the room, and these together enable the occupants of the room to dissipate the heat produced by their metabolisms at the rate at which the heat is produced within their bodies.

A brief illustration, with approximate calculations, of the operation of heating panels and cooling panels may be of interest.

Assume a room of 100 sq ft and 20 ft high; the floor covered with ice, as in a skating rink; the ceiling surface temperature maintained at 100 F, and the walls having a mean interior surface temperature of 80 F.

The ceiling will be a heating surface; the floor will be a cooling surface; the four walls may be heating surfaces or cooling surfaces or neutral surfaces, depending on the relative temperatures of the floor and the ceiling.

The heated ceiling will radiate energy at the rate of 153 Btuh per square feet; 70 percent will be intercepted by the floor and 30 percent by the four walls. The ice-covered floor will radiate energy at the rate of 91 Btuh per square foot; 70 percent will be intercepted by the ceiling and 30 percent by the four walls.

The mean surface temperature of the floor and the four walls will be 53.3 F and the flow of heat from the ceiling to the room will be at the rate of 450,000 Btuh by radiation and at the rate of 645,000 Btuh by radiation and convection, combined.

The mean surface temperature of the ceiling and the four walls will be 91.1 F and the flow of heat from the room to the ice-covered floor will be at the rate of 527,000 Btuh by radiation and at the rate of 750,000 Btuh by radiation and convection, combined.

Since the ice-covered floor takes more heat out of the room than the room receives from the heated ceiling, it follows that the four walls must be heating panels and that heat must flow through them from the outside at the rate of 105,000 Btuh, or at the rate of 13 Btuh per square foot, in order that the mean interior wall surface temperature will remain at 80 F.

To check this calculation: the mean surface temperature of the floor, the ceiling and three walls is 69.5 F; the flow of heat from the wall to the room is, therefore, at the rate of 10.6 Btuh per square foot; this is 82 percent of the total required flow of heat calculated and checks the former calculation satisfactorily.

The mean surface temperature of the floor, the ceiling, and the four walls will be 70 F and the mean air temperature will then probably be about 65 F. When a room is heated by means of panels, the mean air temperature within the room is generally a few degrees lower than the mean temperature of the surfaces enclosing the room.

S. R. Lewis, Chicago, Ill. (Written): I am as happy to contribute my ideas to the Forum as I am unhappy that I cannot be present at the Forum. I have reached no conclusions about any of the methods of radiant heating.

1. I think that no engineer should reach a *conclusion* about anything except that he should conclude that it is unwise to reach a conclusion about engineering applications.

2. The earliest installations of radiant heating designed by myself did use warm air blown under masonry floor construction prior to 1924, as in the Wesleyan gymnasium at Bloomington, Ill., about 1915, and in the basements of several public school buildings in Columbus, Ohio. I employed warm air for radiant heating in the floors throughout the entire Jones Junior High School in Toledo, Ohio, about 1925. I would employ it today in any plant where the conditions were favorable.

3. I have under way at present the conversion of one of the first large general office buildings having but one principal floor in which steam pipes under the floor were installed in about 1938 for radiant heating. The steam pipes, even though supposed to be capable of operation at high temperature, caused so much complaint from too-hot spots that they were abandoned and had to be supplemented by indirect warm air delivered through conventional grilles. We will substitute relatively low temperature hot water in the former steam coils.

4. It is convenient and economical to employ hot water in overhead serpentine pipe coils in multistory buildings such as hospitals, of which we have had two large ones in successful operating experience.

5. We have designed, and found eminently satisfactory, underfloor hot water pipes in large assembly rooms such as places of worship.

6. We have had operating experience with about 30 residences, using overhead hot water pipes imbedded in the plaster, also with steel ceilings. In these we have used ferrous piping and copper tubing.

7. We have designed and expect to have installed a large residence plant using lead-sheathed low temperature electric heating cable imbedded in the plaster.

8. We have employed metal and plaster and even wood facing for hot water radiant heat panels in side walls as auxiliaries to floor and ceiling surfaces as in small bath rooms in residences.

9. I suggest that we have inherited a tolerance of radiant heat coming downward from the sun, and I submit that the unquestionable comfort and success in the plants that have the heat transmitters overhead might point to avoidance of the apparently unnecessary long-time education of the human animal to accept heat rising from underfoot.

10. Our bodily heating system uses a warm liquid in tubes, circulated mechanically. This heating system has been reasonably successful. There is no extended use of a warm gas such as steam or air as a heat carrier in the animal world. The liquid heat carrier seems to lend itself well to control and to durability.

I intend to continue advocating the use of heat transmission via large areas of comparatively low temperature.

R. K. Becker, Evansville, Ind.: Although panel heating or radiant heating seems a complex subject we are still dealing with the fundamental problem of replacing heat losses, and the methods outlined in The Guide for calculating heat losses from rooms still remain in force.

In any panel heating system, equilibrium is sought. As Dr. Winslow pointed out, the air temperature and the surface temperature will seek a balance within narrow limits.

Those who have designed and installed a number of heating panel applications have established limiting factors in design. In my own design, using ceiling panels or sidewall panels heated with warm air, I have limited myself on any ceiling or wall surface to a factor of 70 Btu (per sq ft) (hr. of heat transfer).

In other localities such as Montreal higher factors are successfully used. However, undesirable conditions might result.

Literature regarding classification of structures is needed, for commercial buildings and residences present very different problems. Recommended heat dissipation factors would also help in the matter of design.

E. H. LLOYD, Washington, D. C.: My first question is directed to Dr. Winslow. What provisions for ventilation requirement should be made in conjunction with radiant heated spaces? You recommend a minimum air movement within a radiant heated space. I can cite two examples where, following the installation of radiant heating systems, ventilation had to be added.

The second question is that Mr. Boester elaborate on the type of tubing coils he has used, particularly as to the size of tubing and method of application to plaster or other medium used as a panel.

DR. WINSLOW: With regard to the question of air change, the suggestions in THE GUIDE may be followed. Air is changed to minimize the accumulation of odors, which can be done by a change of about 10 cfm per person. Whether artificial convection is needed to accomplish the air change depends on the methods of construction. With normal methods, a desired amount of interchange will take place around windows and doors and even through structures. With the more hermetically-sealed types of construction, it may be necessary to make provision for the minimum amount of air change.

MR. BOESTER: Tubing, which is preferred because it permits a type of panel that has low thermal capacity, generally is available in smaller sizes. We have experimented from 1 in. down to ¼ in. The small-diameter tube is desirable and the easiest way to apply it is by stapling. In the case of a ceiling with wood joists, and gypsum lath, the tubing is stapled wherever it crosses the joist. In the case of a metal lath ceiling, the tubing could be wire-tied to the lath.

Regarding plaster, if fiber gypsum is used, plaster should first of all be applied the conventional way. For ¼-in. tubing, the total plaster depth would be about ⅜ in.

If the heat tiering is properly worked out, the small tubing does not present any problem of pressure drop.

C. F. MALLY, Detroit: In this discussion I should like to contribute a description of a heating installation in Detroit. The building, a Quonset hut constructed with strand steel joist, was to be heated with air. State laws made it necessary to keep the heating system outside the building and yet close enough so that the ducts would not have to be insulated.

The building was a nursery school, 110 by 20 ft with no form of wet heat permitted. According to THE GUIDE, with this great heat loss only 70 F temperature could be obtained inside with 39 F outside.

We designed a system employing air flowing underneath 2-in. porous slabs. A perforated baseboard at each end of the room allowed for fresh air.

We found that the temperature of the floor was about 86 deg maximum, but never over 80 F under normal conditions. The potential thermostats modulated the amount of air coming in through the chambers of each end of the building, thus maintaining a temperature that was very accurate.

Fresh air was continuously taken in from the outside to give about 30 percent of fresh air in the room, with about 5 percent leakage through the dampers.

RALPH POOLE, London, England: I should like to invite the A.S.H.V.E. to visit England to see one of the best examples of radiant heating, traced back to Julius Caesar, who tried to colonize Britain but failed because his men could not endure

the English climate. Many years later a second expedition left behind some of the finest examples of panel or radiant heating that you can find. In St. Albans are two Roman buildings containing radiant heating, one a house and the other a bath, parts of which are in a good state of preservation. Thus, it is surprising to hear Americans giving the credit for panel heating to an Englishman of modern times.

Our research has shown that radiant heating as applied to industrial problems must be planned with care. In industries such as cotton and rayon spinning, it is necessary to operate at high humidities. Because water vapor absorbs much radiant heat, conditions in a textile mill would be different from those in the relatively dry atmosphere of houses and offices.

These conditions are particularly marked in cotton mills doing fine count spinning. Conscious of the effect of the radiation from their bodies to the machines, the operators often complain of feeling cold because the machines are too cold. Because of high humidity, the heat is absorbed in the atmosphere before it reaches the machine. Hence, in order to give comfort in fine spinning, the rooms are operated at very high temperatures thus reducing the radiation from the operator's body to the surrounding atmosphere and to the adjacent machinery. This is a point to be stressed when we pass from ordinary offices into industrial places of high humidity.

H. C. Sharp, St. Louis: Mr. Angus, does apparatus exist today for 100 percent transfer of heat by radiation?

Mr. Angus: I do not think it does, and in the ordinary building there is no panel heating and radiant heating. Although in some special case, all the heat could be transferred by radiation, most of the heating systems are a combination of radiant heating and convection heating.

F. W. Legler, Minneapolis: I have in my hand pages 46 and 47 from the December issue of *Better Homes and Gardens* entitled, "What is Radiant Heating?"

The article claims that with radiant heating, more comfort can be secured for less money. This article, written by Bob Jones and H. A. Holbrook refers to homes.

A statement is made in the article that "unclothed subjects can be comfortable in 50 F air temperature if walls, ceilings, and floors are warm." I would like to see that demonstrated. The article also states that savings in operating costs run as high as one-third. Just where Messrs. Jones and Holbrook got this information I don't know.

I think the Society should be aware of the wild statements which are contained in numerous magazine articles. This is not the only one which is incorrect; there have been many published. I would like to see in these magazines information which might originate with the Society.

Dr. Winslow: I would like to assure the speaker that the statement about 50 deg is entirely correct. That was quoted from some of our experiments at the Pierce Laboratory. It is true that a nude subject is perfectly comfortable with air at 50 deg and completely surrounded with walls of 110. We maintained those extreme conditions in order to work out fundamental laws involved, and they have, of course, no practical application.

I am glad the matter is brought up because some years ago the noted advocates of panel heating gave the general public the impression we were going to live in rooms of 50 and walls of 110. Marked contrast can be produced only by extremely artificial and costly experimental installations.

Regarding savings, I entirely concur with what has been said. The only theoretical possibility of economic saving is in a building with very high room ceilings, in which it is possible to have some economy by avoiding the concentration of hot air near the ceiling.

Mr. Morrison: In regard to fuel savings, assume that one who has lived in a house of certain size, builds another 25 percent larger, with radiant heating, insulated walls and ceilings, and double glazed windows. The new house will probably have

70 percent of the heat loss of the previous house, while the new heating plant will be 25 or 30 percent more efficient than that in the original house. After a year the house owner will have possibly 50 percent of the heating cost of the smaller house. Consequently, sometimes more credit is given to the panel heating system than is warranted.

ROBERT GREENE, La Porte, Ind.: I have three questions which perhaps could best be answered by Dr. Winslow. What humidity conditions are desirable in these heating systems? Is the addition of moisture required? If so, how is this accomplished?

DR. WINSLOW: I don't think humidity enters into the problem at all. In the limits of any occupied space (I am not talking of industrial processes) variations occur when it is hot and when little humidity is desired. The variations in humidity at comfort temperature and below comfort temperature are insignificant in their physiological effect. That is a point emphasized in Professor Rowley's recent work and in the criticisim of the A.S.H.V.E. "Comfort Chart" which I think will have to be revised from the viewpoint that it gives too much emphasis to relative humidity in the low range. In uncomfortably hot conditions, humidity is vitally important.

H. M. HART, Chicago: In the structure in which the heat loss was greater through the walls, what type of control were employed?

MR. BYERS: To control temperature in a sheet steel building heated by heavy concrete slabs, if a good comfortable condition and straightline temperature control are desired, the slabs will not respond as rapidly as the inside temperature will respond to outside temperature. It is a difficult problem.

MR. HART: If the structure of a residence contains a normal amount of glass, i. e., single glass and not double, and a floor panel system of heating in concrete slab, I think the lag in the floor will be greater than it will be in the walls. I don't believe an outdoor anticipating thermometer will respond and maintain a uniform temperature in the house. That has been my observation on one installation.

MR. BYERS: That observation follows the theory on which I was basing my assumption on control. In other words, the slab should not be used in a house having a lag which is short for the structure. An outdoor controller can be used to sense it immediately, but if the floor will not respond quickly enough to meet the requirements of the building, a drop in temperature will result.

When it is a question of a long lag or a short lag, we have yet to know definitely what the time lag of all these structures is.

J. E. HAINES, Minneapolis: The question of heavy panel in the floor as compared to a frame structure has been under examination for some time. On a change of a demand of say 10 percent a normal floor panel of the concrete type will require about three hours, by test and computation, to reach its new equilibrium, *i. e.,* 10 percent less heat output. While in the case of a frame building, if the sun comes out from behind clouds it may be 10 minutes for that 10 percent to be felt completely inside the building. Thus, a comparison of 10 minutes to 3 hrs is something that no control system can possibly overcome with an inside thermostat or outside thermostat.

I would like to reemphasize that comfort can be obtained only by adequate practical control. Control should therefore be considered at the beginning of the design of any heating system, and particularly in radiant panel heating, where the system itself is directly over the head or under the feet of the occupants. In no case should the thermal characteristics of the panel be greater than the thermal characteristics of the building.

LESTER T. AVERY, Cleveland, Ohio: Some years ago Professor Giesecke pointed out that panel cooling should not be overlooked as is the case most of the time. Now I am not thinking of residential heating, but of heating in offices, restaurants, stores—where there is a heat gain inside which is greater than the wall loss. To try to accomplish tempering of the wall by applying a heating coil in a floor is doomed to failure.

I think Mr. Legler mentioned a tendency which engineers as well as publications must avoid—that is, proceed from erroneous assumptions to foregone conclusions.

If you can visualize this room as being cooled by a stratum of air that comes across at that level, which as Dr. Giesecke says is cooling the ceiling, we have in effect almost panel cooling. The ultimate of that panel system comes with the perforated ceiling construction distributing the air.

I agree with Mr. Haines, who says that though control is needed, it can't be expected to do the impossible. With air distribution, temperature can be quickly changed. This room can be changed from a heating problem to a cooling problem, with 500 people coming in. If the ceiling is uniformly cool, it can be done with less cool air than if we have spot grilles introducing cold air.

That is a function, a result of the panel method of air distribution. Whether it is called heating or cooling, it should not be overlooked. The building we spoke of heating will eventually need cooling, and ventilation cannot be disregarded. The question is raised, How much air for ventilation?

If a very broad ceiling area is utilized, if advantage is taken of lower temperatures on ceilings and if the air is distributed uniformly, the problem is simplified and the controls will work.

W. J. WADSWORTH, Cleveland, Ohio: The discussion of the tin building that Mr. Byers began would lead one to believe that the application of panel heating even as carried over to a home installation would not be very satisfactory because of the fact that the panel would have a much longer time lag than the wall of the structure. I should like to have Mr. Byers discuss that and perhaps indicate whether he thinks that type of installation would or would not be satisfactory for an ordinary small home.

MR. BYERS: The statement that wide fluctuations in temperature would result is probably only true when a very rapid drop in outside temperature occurs. Normally, the fluctuation might follow the outside temperature drop closely enough. On the other hand, temperature control in many homes is not perfect. Two people may not find equal comfort in a house.

I am glad this question came up because I know that many homes are being built with concrete floor slabs and wood frame structures, and with which the occupants are very pleased. Now, do these homes have good control? Time is the best teacher.

In my preliminary remarks, I tried to keep from mentioning any individual form of heating medium, as water, air or electricity. I tried to talk about heat flow, heat emission, and any kind of heating surface. My example of a heavy slab which had a long time lag, is applicable to any form of heating medium.

W. L. FLEISHER, New York: Heating is a particular kind of comfort which is involved with higher temperatures. In the example Dr. Winslow used of walls being 50 F even though the average temperature was 68 F, I do not believe the occupant would be comfortable with his back against the 55 deg wall. Perhaps the temperature in the middle of the room would be comfortable if one kept away from the walls and windows, but there would be no opportunity to vary those conditions. In a house, particularly one heated by means of pipes in the floor, sleeping temperatures as contrasted with rising temperatures would be a problem.

I wish to contradict Dr. Winslow on the question of humidity which he and I have discussed many times before.

At a meeting of 500 medical men, Dr. DuBois said that in over 25 years he had not been able to discover from a physiological angle that his subjects had shown any differences with different amounts of moisture in the air. I asked him whether he had examined any of his subjects from a psychological angle, because I thought I could bring many people into any enclosure in the winter who, if the humidity was very low, would be uncomfortable or who would have electrical discharge in their fingertips and would ask for humidity.

I disagree on the point that moisture in wintertime is unimportant and I think it is a great assumption on Dr. Winslow's part to say so.

DR. WINSLOW: Mr. Fleisher has made some tall assumptions. He did, however, make two points of the greatest importance. A method of supplying quickly responsive panel heating is vital, but as Mr. Byers pointed out, there is no reason why it shouldn't be done.

It is rather unfortunate perhaps that we have come to associate panel heating in this country with the particular form which involves the use of concrete and other substances of very high heat capacity. That is by no means necessary. There are many ways in which surfaces of quick response can be heated and used as panel surfaces.

Then, the other point he made about the window I want to reemphasize. As I suggested in my introductory remarks I should like to know very much if any of the designing engineers in this field have yet attacked this problem. I think if panel heating is to succeed, it is essential that it should be applied just as convective heat is applied at different rates in different parts of the room. There will be no difficulty at all in designing a panel so that the radiant temperatures near the window are higher than those at the other end of the room. I think this is a subject that the designing engineer must attack. I should like to know how far have the heating panels been varied in number or temperature so that excessive radiation coming from the window will be balanced.

MR. POOLE: Perhaps I did not make it clear that Julius Caesar did not stay in Great Britain because his engineers could not correct our climate.

I would also like to say that there is great danger in getting standardized conditions. I wonder if we want to get the conditions in buildings, rooms and everywhere we go just the same. I do think that the idea of getting an absolute pure atmosphere that is always the same is not so good. I think the medical profession would tell us that the change from one temperature to another stimulates life, so that we think twice about getting our controls too perfect.

May I make one other point which has not been brought out. One good thing about radiant heating is that the rays do their job without making any noise. To ventilate noiselessly with moving air is difficult. In libraries for example where things often get musty, silence is an important requirement. It we can solve the problem with adequate panel or radiant heating we will have achieved something.

A. A. ENGELBACH, Washington, D. C.: In my experience with public housing during the war, I became interested in floor panel heating. In connection with housing, we have constructed several panel heating systems using concrete, and one system having pipes under a 1½ in. wood floor.

In general, these projects have performed satisfactorily for about two years. Throughout the past two winters the temperature has perhaps never gone below 10 deg above zero. The Washington temperatures as a whole average around 40 F. In my opinion the floor panel design, held to 85 F, is erroneous. The panel may be safely designed in the southern latitudes of the United States (where the temperature is as high as 40 F average) to a design temperature of 100 or 102 F with a much higher output of Btu than is at present suggested because of the fact that there is a psychological reaction felt by anyone on a cold day.

CHAIRMAN KAYAN: Is it your idea that floor panel temperature should be higher, Mr. Morrison?

MR. MORRISON: Mr. Engelbach brings out the point that possibly we have been designing our panels for too low a temperature on floor panels. Possibly the flow of heat from the floor panels is greater than published data now give, that is, the information that the published data give us is possibly on the lower or safe side. My own observations have been that temperatures in the neighborhood of 85 F on floor panels are uncomfortable. I think that possibly some of the factors we are using

may be a little on the safe side. But I do disagree with Mr. Engelbach in that you can go above 85 F. I don't think that it would be comfortable.

Now, it might be permissible to occasionally go above that temperature during maximum conditions. In other words, you design for a certain temperature which may be realized only once in 5 or 10 years. The people might be willing to put up with high floor temperatures for short periods in order to get the type of system they like. Panel heating has seized the public fancy as air conditioning did 10 or 12 years ago, and we should avoid the installation of untested and untried systems.

Mark Jenks, Los Angeles: Regarding Dr. Winslow's question on developmental work on prefabricated panels, I have had some experience in installing heating coils at the intersection of the wall and the ceiling. We have developed a low-cost boiler, and insulation has also been taken into account. The panel is made of copper pipe, with fins, and can be either imbedded or exposed.

S. Konzo, Urbana: Regarding the difference between the floor surface temperature as calculated and the temperature as experienced in practice, in both the warm air residence and the I=B=R Research Home, the following observation may account in large measure for this apparent discrepancy.

If the design heat load of a house is 50,000 Btu per hour, under say, minus 10 conditions, and we assume that the delivery at the radiator or at the registers should be equal to 50,000 Btu per hour, we also assume that we will have a boiler output that is considerably higher. In other words, the difference between the boiler output or furnace capacity, and the radiator or register delivery is assumed to be the piping loss.

In an actual installation the piping loss serves to heat the house. A regain of heat is obtained from boiler casing, smoke pipe and chimney that is of considerable magnitude. This magnitude may be as large as 25 to 50 percent. Hence, for the sample house quoted, only about 25,000 to 40,000 Btu per hour are given up directly by the radiator or register and the remainder comes from indirect sources. As a result it has been found that the actual operating temperatures for a hot water or warm air system will be considerably lower than the values assumed in the design.

Mr. Morrison: Concerning heat methods for a heating system, the usual practice, is to allow for the heat regained from the boiler warm-air unit breeching, and piping and neglect to put any heat in the utility room. Most of the houses using the floor panel heating system have a utility room and laundry room. Usually, the designing engineer will take into account the heat regained and will either make provision for shutting off any heating units that might be installed in the boiler room, furnace room or basement or he will neglect to provide any heating there.

W. F. Ryan, Salina, Kansas: I would like to ask two questions of Mr. Boester. First, How much research has been done on panel cooling and what are the temperatures available in the panel? Second, How near can we approach the dew-point temperature in the room in the structure mentioned?

Mr. Boester: We have not done any panel cooling, as yet. We are planning several projects but we have not had any actual experience.

Vic Sanders, Pittsburgh: Some years ago, in supplying fuel to a large city 100 miles away from the source of supply, we were confronted with a temperature drop of 10 to 15 deg and a time lag of about 5 hrs. However, our real concern was the possibility of snow or rain. Has the condition been studied regarding its psychological or other effect?

Mr. Morrison: In Oregon, some research revealed that heat loss through wetted walls was considerably greater. But that doesn't answer the question of snow, only of rain.

G. Lorne Wiggs, Montreal, Que., Canada: In Canada, hot water heating, the most common heating system there, owes a great deal to the works of Thomas Tredgold and Charles Hood, both of England. While many copies of Tredgold's book found

their way into Canada and the United States, undoubtedly most Canadian hot water heating practice was developed from Hood's work, which by 1894 had reached seven editions.

At one time one of the outstanding authorities on hot water heating in the Society was the late A. W. Moulder, under whom I had the privilege of working for a short time. Mr. Moulder was a great disciple of Prof. A. H. Barker of England.

With reference to some of the remarks made here today, it is time, I think, for the Society to perform the necessary research to correlate the present A.S.H.V.E. Comfort Chart with the mean radiant temperatures of the floor, walls and ceilings of the enclosures. I have a suspicion that the difference between the so-called summer comfort zone and the winter comfort zone is simply a difference of the mean radiant temperature of the enclosure. Most of us interpret radiant heating in the light of our knowledge of and experience with convection heating but in order that radiant heating systems be properly designed and controlled it is important to have an authoritative chart showing how the human body responds to variations in the mean radiant temperature and the air temperatures of the enclosure. It would be well also, to know whether the human body is affected more by variations in the mean radiant temperature than by changes in air temperature.

Dr. Winslow asked a question about people working close to the windows of radiant heated buildings. In reply to that I would like to say that two years ago we designed a textile mill which had about 10½ miles or more of pipe buried in the concrete floors and ceilings. We installed the radiant heating in the floor slab of the lower floor and in the ceiling slab of the upper floor. The interesting thing is that even at the outdoor temperatures which we get in Montreal, down to —30 F or lower, the operators can work practically with their backs right up against the windows and not suffer the discomfort they would with other forms of heating.

PAUL S. PARK, JR., Pittsburgh: Mr. Boester stated that the ceiling system is superior to a floor system for a number of reasons, one of them being that one square foot of ceiling surface will do the same job as two square feet of floor space. If this is correct, I assume that we can use in a ceiling job approximately 50 percent of the coil surface used in a floor job, and get the same performance.

It was also stated that ceiling installation costs are less than floor installation costs. I should like to say that although opinion varies I have found that installation costs are usually less for ceilings than for floor systems.

MR. BOESTER: Answering your question about performance of ceiling versus floor, it is not a question of using fewer feet of pipe surface. It is the performance of the surface. The ceiling can operate at a higher temperature. I can agree that possibly you can go a little higher with floor temperature than the general consensus feels we should, but normally there are not enough square feet of floor area in the average house.

As far as costs are concerned, there are many floor systems than can compete with the ceiling systems, but it depends on what is put into the ceiling versus what is put into the floors. Though floor systems are in the majority, they are, from my viewpoint and observation, primarily gravity warm air heating systems, since most of the heat occurs by convection rather than by radiation. There are many hazards of floor coverings, furniture placement and other factors that affect the overall performance.

MR. BYERS: An installation I witnessed was laid out for a constant flow of water to a concrete floor in an office building. The panels were connected to a complete reverse return system. We were going to modulate the steam to the converter by a room thermostat.

On visiting the job, I found that a self contained steam control valve had been installed to provide a constant water temperature to the floor coils. The pump was controlled from a room thermostat. The pump had to run continuously to get the

system to heat evenly and so the room thermostat was set up to 85 deg and the water temperature was manually set on control valve.

But even though the control valve was 100 ft away, this panel system was considered as satisfactory as any other system.

M. W. KEYES, Neenah, Wis.: It was stated in an example that a slab has a three-hour lag and the walls of the residence have a 10 min lag, assuming single glass. Nothing was said as to whether or not the walls were insulated or to what extent.

In insulated houses, the type of plaster makes a large difference especially in the summer. The heating lag is different because the plaster has much more mass. Assuming summer conditions, the heat will not get into the house as early in the day if the interior finish is plaster as it will if it is a light-weight building board of some kind.

I would like to ask to what extent an insulated wall and insulated roofs, with perhaps a U factor of 0.12 will increase the heat lag of those walls and roofs. Also let us assume that double glass is applied instead of single glass. Would that come anywhere near the three-hour lag that the slab was said to have and thus allow better heating control?

A. R. CURTIS, Salt Lake City: I am wondering if the gentlemen would direct some discussion relative to using a floor panel plus unit heaters in a cold climate. An example is a freight dock with doors open a good deal of the time and with the personnel working on the floor where higher temperatures might be uncomfortable for them.

A. J. BUCKHORN, St. Louis, Mo.: In my experience I have found the biggest problem to be intermittent circulation and continuous circulation. The thickness of the floors also plays a part in heating efficiency. I have found over a period of years that there is a wide difference between theoretical studies and field practice.

I am wondering what can be done to help the members of the Society toward getting a concrete example or a guide to follow for low-cost installation, and solve such questions as that of continuous circulation or intermittent circulation.

MR. MORRISON: The question that was brought up by Mr. Curtis concerns the use of radiant panel heating in connection with unit heaters on a freight loading dock. The use of floor panel heating alone would heat the building as long as the doors were closed. But with the doors open, unit heaters could attempt to blanket the doorways as much as possible. At best, it is difficult to do this but the amount of draft from the door can be cut down.

FERDINAND JEHLE, Indianapolis: One of the speakers mentioned a floor temperature of 102 F, which he thought was comfortable. I believe Dr. Winslow could give us figures on the maximum temperature that the human being can stand on his feet.

I believe that 102 F may produce, not physical comfort, but psychological satisfaction with one's heating plant. That is, if even at the lowest temperature you can get so hot, that may produce some satisfaction.

That brings up another question which may have been discussed before I came in. To produce comfort, I understand that the room or the air temperature should decrease with the outdoor temperature. I question if you put an arrangement like that in a man's house whether physiology or psychology will win.

DR. WINSLOW: I know no reason why indoor temperature should vary with outdoor temperature.

As to the floor, we do not have as far as I am aware any accurate or reliable data on maximum floor temperature. I should be very much surprised, however, if in general, people would be comfortable with a floor temperature of over 85 F, but there are no solid data to confirm that.

W. M. WALLACE, II, Durham, N. C.: Is it possible that we can make some investigation regarding the proportionate amount of convection that takes places from the floor for various temperatures? It is my opinion that the convection on the floor

reduces the floor temperature and at the same time gives a greater Btu output. I think it would be in order if the Society could make an investigation on that point.

Nothing has been said regarding the calculations of rooms having very high ceilings, such as 40 to 60 ft high. If you take into consideration all of the losses under the normal Btu calculations of these high rooms it is impossible to correlate the figures for panel or floor heating. However, it appears from some installations that satisfactory results can be obtained although they do not check with the calculations.

Mr. Boester: I would like to comment on the suggestion concerning the floor study. I believe Professor Rowley of the University of Minnesota is now getting ready to conduct some very interesting studies along that line. I do know he is setting up equipment for the purpose of definite control of the other five surfaces of the test space. He might be persuaded to give some study to the problem of the floors just mentioned, and also the matter of foot comfort.

D. L. Mills, Rome, N. Y.: I would like to ask Mr. Angus if a distinction might be made between radiant heating and panel heating on the basis of areas and intensities, by which a radiant heating system might be one in which the heat source is comparatively small in area and high in intensity and a panel heating system might be one in which the heat source is rather large in area and rather low in intensity.

Mr. Angus: I agree that it would be a good idea if a distinction were to be made. Actually, panel heating covers a great part of the room. We have isolated cases where there isn't much exposure; a high temperature panel could be used to heat the space.

Mr. Keyes: My original question was: To what extent will double glass, insulated walls and insulated rooms increase the time lag of the walls, roof and windows to somewhere nearly equal the time lag of a concrete slab used as a heating source?

From what I have observed, if the time lag of the walls, etc., is assumed to be 10 min, as compared with three hours for a concrete slab, then double windows, insulated walls and insulated roofs would change that lag to something like one and a half or two hours for the walls and roof, *i. e.*, approaching the time lag of the slab.

P. S. Morton, Lawrence, Michigan: I would like to ask what effect complete coverage of the floor with thick carpet has on the output.

Mr. Morrison: I have some observations indicating that the reduction in heat flow through the carpet is not as much as we might think. If a carpet surface is put on a concrete floor, it is true that the floor has been insulated but at the same time the radiating surface of the floor has been increased by an immense number of wool fibers. Thus the reduction in heat transfer through a carpet is not as bad as might be assumed at first.

Mr. Boester: I concur in Mr. Morrison's reasoning. As far as laboratory tests are concerned, about every carpet used would have to be tested because the fiber structure is entirely different. Some testing organization should set up facilities for very extensive testing of the fibers involved for flooring, particularly of the rug type.

S. J. Heiman, St. Louis: In the past several years I have tried various ways of heating small homes. In one case, warm air was introduced into a concrete slab by means of a sewer tile right in the slab and the sewer tile at the perimeter of the house, with grilles at each corner of the house. From my experience I conclude that possibly in radiant heating we sometimes have the tendency to introduce heat where it is not needed. In other words, it would be far better if the heat rose at the point of heat loss.

H. M. Nobis, Cleveland: The speaker from St. Louis mentioned 50 percent in fuel savings. What types of buildings were compared and what was the heat input from other sources?

Mr. Buckhorn, St. Louis: In the fuel saving we are building up heat in our floor and can draw on it as needed. On bedroom cooling in my locality the opening of the windows solved the problem without complaints.

MR. MORRISON: In regard to fuel savings, that point is brought out in my introductory talk. You can't make any claim as to fuel savings without having identical houses with identical insulations, with identical people living in them, with identical heating plants set for identical efficiencies. We will have to discount any claims for fuel savings for any type of equipment, because I can think of instances where installation of replacement heating equipment has brought forth savings as much as 50 or 60 percent in the same house with the same people living in it, and that, of course, could be laid only to a difference in efficiency of the two heating plants, not in the type of heating system.

J. R. FELLOWS, Urbana, Ill.: One question that hasn't been answered is that of the proper floor temperatures in houses without basements but with ceiling panels. How do they compare with the low floor temperatures obtained with the conventional or circulation convection heating?

MR. BOESTER: If very careful engineering is applied to the job, and if adequate insulation is used in the walls and above the ceiling panel, the floor temperature and floor air temperature should be reasonably satisfactory.

GORDON HYMAN, St. Louis: In our study of the different types of radiant heating, especially in inexpensive homes, we wish to heat these homes well and yet keep the cost low. In the common concrete slab radiant heating system where control offers such a problem, if we ignore the complete radiant system and instead use a combination radiant system with a circulated air, and keep the floor temperature near 70 F instead of raising it 10 or 15 degrees above that we will have a better all-round heating system.

MR. BOESTER: I would like to say that from a theoretical viewpoint the split system would have many advantages, but I haven't been able to figure out how you could use it economically. I think we have a lot more to learn. Other systems can provide equal comfort if intelligently applied. I think the split system, particularly from the viewpoint of adequate ventilation, is quite desirable.

L. J. HARRINGTON, Portland, Ore.: I found a great tendency among some of the contractors and some of the smaller manufacturers, to do what I have called spot heating. In other words, they rate a surface at so many square feet, and set it in between possibly four stud spaces in an outside wall. We haven't had enough experience as yet to know just what is happening. I wonder if Mr. Boester has had any experience and can tell us just what we get from the standpoint of discoloration of the plaster.

MR. BOESTER: You are now describing a job using the electric elements in the plaster?

MR. HARRINGTON: No, it might be just pipe grids that fit in between the 16-in. center studded spaces, say 4 of them in a room.

MR. BOESTER: We have not observed it.

MR. HARRINGTON: Have you had anything that would indicate whether we should get discoloration from partly covered ceilings?

MR. BOESTER: The discoloration comes from a number of things. Discoloration, of course, is a matter of kind of paint or covering. Discoloration is the temperature and also, you might say the electrostatic condition. A lot of discoloration is caused by air movement, in other words, accelerated air movement across the heated surface, with the result that the dirt in the room is in circulation and deposition of that dirt takes place. It isn't an actual physical discoloration occurring from under the surface.

L. K. REISBERG, Minneapolis: Mr. Byers mentioned the self-leveling item of a floor slab. Once I observed the performance of a panel heating system during a 45 to 50 deg drop in outside temperature. Although boiler water temperature had not been increased during the temperature drop, the temperature inside the building was only one degree lower than it had been the day before.

The fact has not been brought out that as the room temperature drops, automatically the capacity of the slab is increased by the greater difference in temperature between

the slab and the air and mean radiant temperature. But with the same boiler water temperature and with a 50 deg difference outside temperature the temperature inside remained constant.

Professor Kayan thanked all those who had taken part in the discussion which he believed had been quite instructive to the audience.

Vice Pres. G. L. Tuve then called upon Art Theobald, Beverly Hills, president of Southern California Chapter, who reported that Southern California Chapter had extended an official invitation to the Society to hold its summer meeting at the Hotel del Coronado which was located across the bay from San Diego, Calif., and that this invitation had been accepted for the dates June 1-4, 1947.

Mr. Theobald remarked that those attending the meeting would have an opportunity also to take part in the meeting of the *American Society of Refrigerating Engineers* from June 9-11 at Los Angeles.

The meeting adjourned at 5:15 p.m. after giving a rising vote of thanks to Messrs. Kayan, Angus, Boester, Winslow, Morrison and Byers for the expert advice which they had provided during the meeting.

FOURTH SESSION—WEDNESDAY, JANUARY 29, 9:30 A.M.

1st Vice Pres. Baldwin M. Woods, Berkeley, Calif., called the meeting to order. In his opening remarks he referred to the past collaboration of the Society and the medical profession and with others having an interest in the problems of public health. It was, therefore, quite fitting, he said, that the session was to be devoted to a presentation of contributions by men working in the field of medicine and public health. Three papers were given and considerable discussion followed. The fourth session adjourned at 12:10 p.m.

FIFTH SESSION—THURSDAY, JANUARY 30, 9:30 A.M.

Pres. Alfred J. Offner, New York, N. Y., called the meeting to order at 9:30 a.m. in the Euclid Ballroom of the Hotel Statler.

Resolutions

G. D. Winans, Detroit, Mich., presented the following report of the Resolutions Committee and moved the adoption of the report, which was then adopted by unanimous vote:

WHEREAS, Alfred J. Offner has served on many Society committees as member and chairman, has served on the Council since 1935 to the present time, was Treasurer from 1935 to 1938, was Second Vice President in 1944, First Vice President in 1945, and has been President of the Society in the year just ended,

WHEREAS, His continuous and able application to Society duties has been outstanding and has contributed to maintaining the high standards set by previous officers,

BE IT RESOLVED, That the A.S.H.V.E. extends its grateful thanks and appreciation to him and to the gracious Mrs. Offner who accompanied him on many of his travels to Annual, Semi-Annual and Local Chapter Meetings.

* * *

WHEREAS, The health of L. P. Saunders, Chairman of Committee on Research, has not permitted him to be present at this meeting, and

WHEREAS, His efforts on behalf of the Society's Research Program and the Society's welfare, have been great and untiring, therefore

BE IT RESOLVED, that the A.S.H.V.E. thanks Mr. Saunders for his many contributions and wishes his speedy recovery.

* * *

WHEREAS, the 53rd Annual Meeting of the Society has been an exceptional success due to the efforts of the Northern Ohio Chapter through its Committee on Arrangements under the able leadership of D. L. Taze, and

* * *

WHEREAS, many organizations and individuals have participated in providing members and guests with opportunities for instruction and entertainment, therefore,

BE IT RESOLVED, That an expression of appreciation be adopted and that copies be sent to each of the following:

To D. L. Taze, Chairman of the Committee on Arrangements and to his several committee members and their ladies.

To Dr. A. C. Willard for his address at the dedication of the Research Laboratory.

To the authors of the technical papers.

To Prof. C. F. Kayan and his panel in discussing panel or radiant heating.

* * *

To L. T. Avery, toastmaster of the banquet.

To C. F. Roth and his staff for the splendid 7th International Heating and Ventilating Exposition held concurrently with our meetings.

To Dr. Kirtley Mather for his address at the annual banquet.

To the Cleveland hotels for their cooperation in a difficult situation and to Hotel Statler for an excellent banquet.

To the newspapers and trade papers for their coverage of this annual meeting.

To the Chapters and individual members who have contributed so generously toward the retirement of the mortgage on the Research Laboratory.

Respectfully submitted,

G. D. WINANS, *Chairman*
W. BRUCE MORRISON
C. ROLLINS GARDNER

Installation of Officers

President Offner announced that the next order of business would be the installation of the Society officers for the year 1947 and requested that Past President W. T. Jones, Boston, conduct the installation.

The following officers were then conducted to the rostrum and installed by Mr. Jones assisted by Dr. A. C. Willard, Prof. G. L. Larson, W. H. Driscoll, E. Holt Gurney: *President,* Baldwin M. Woods, Berkeley, Calif.; *1st Vice President,* G. L. Tuve, Cleveland, Ohio; *2nd Vice President,* Alfred E. Stacey, Jr., Syracuse, N. Y.; *Treasurer,* John F. Collins, Jr., Pittsburgh, Pa.; *Members of Council (three-year term):* M. W. Bishop, Milwaukee, Wis.; C. F. Boester, Lafayette, Ind.; Leo Hungerford, Los Angeles, Calif.; Reg F. Taylor, Houston, Tex.; and E. N. McDonnell, Chicago.

As there was no further business, the fifth session was adjourned with the announcement that a joint session with the *National Warm Air Heating and Air Conditioning Association* would be held in the Public Auditorium at 2:30 p.m.

SIXTH SESSION—THURSDAY, JANUARY 30, 2:30 P.M.

The sixth session, a joint meeting with the *NWAHACA,* was called to order at 2:30 p.m. in the ballroom of the Public Auditorium, by Alfred J. Offner, representing Dr. Woods, the newly installed president of A.S.H.V.E. Mr. Offner brought the greetings of the Society to the *NWAHACA* and extended an invitation to those present to visit the Society's Research Laboratory which was open for inspection until January 31.

F. E. Mehrings, Peoria, Ill., president of the *NWAHACA*, as co-chairman of the session, extended the greetings of that association to the A.S.H.V.E. He spoke of the close cooperation of the two organizations in the past and referred particularly to participation of association members in the exposition conducted under the auspices of the Society.

Chairman Mehrings introduced Prof. S. Konzo, Urbana, Ill., who presented his paper.

Dr. C.-E. A. Winslow assumed the chair and declared the meeting open for discussion of the paper.

Chairman Winslow thanked Professor Konzo for his presentation and as there was no further business, the 53rd Annual Meeting was adjourned at 4:30 p.m.

PROGRAM 53rd ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

HOTEL STATLER JANUARY 27-31, 1947 CLEVELAND, OHIO

Sunday, January 26

1:00 P.M. REGISTRATION (Hotel Statler—Mezzanine Floor)

1:30 P.M. Council Meeting (Hotel Statler—Parlor C)

Monday, January 27

8:30 A.M. REGISTRATION (Hotel Statler—Mezzanine Floor)

10:00 A.M. OPENING SESSION (Research Laboratory)—

Welcome by J. E. Wilhelm, President of Northern Ohio Chapter

Response by President Alfred J. Offner

Dedication of Research Laboratory—By Dr. Arthur C. Willard

Reports of Officers

Nature of Air Flow at Suction Openings by A. D. Brandt, R. J. Steffy and R. G. Huebscher

Report of Committee on Constitution and By-Laws by W. T. Jones

Report of Tellers of Election by Walter Sherbrooke

12:00 NOON Research Luncheon (Quad Hall Restaurant—7500 Euclid Ave.) Tickets at Registration Desk

12:45 P.M. INSPECTION TRIPS

Research Laboratory

Nela Park Lighting Institute and A. G. A. Laboratory

2:00 P.M. Opening of Exposition (Lakeside Hall)

9:30 P.M. Operation Relaxation (Hotel Statler—Grand Ballroom)

Informal Musical and Comedy Program with One Act Play by Chagrin Valley Little Theatre

Tuesday, January 28

8:30 A.M. REGISTRATION (Hotel Statler—Mezzanine Floor)

9:30 A.M. TECHNICAL SESSION (Hotel Statler—Grand Ballroom)
The Effects of Moisture Content on the Diffusion of Odors in the Air by Richard L. Kuehner
Dehumidification—Methods and Applications by John Everetts, Jr.
Rating Dynamic Dehumidification Equipment by E. R. Queer and E. R. McLaughlin

12:00 NOON Exposition (Lakeside Hall)

12:15 P.M. Luncheon—Guide Publication Committee (Hotel Statler)

2:00 P.M. PANEL AND RADIANT HEATING FORUM (Hotel Statler—Grand Ballroom)
Professor Carl F. Kayan, New York, N. Y., *Chairman*
1. IS THERE A DIFFERENCE BETWEEN PANEL HEATING AND RADIANT HEATING? Introduction by H. H. Angus, Toronto.
2. WHAT HEATING MEDIUMS FOR PANEL AND RADIANT HEATING SYSTEMS? Introduction by Carl F. Boester, Lafayette, Ind.
3. WHAT IS THE COMFORT TEMPERATURE IN PANEL HEATED ROOMS? Introduction by Dr. C.-E. A. Winslow, New Haven, Conn.
4. DOES A PANEL OR RADIANT HEATING SYSTEM SAVE FUEL? Introduction by W. Bruce Morrison, Portland, Oregon.
5. CONTROL REQUIREMENTS FOR PANEL OR RADIANT HEATING SYSTEMS. Introduction by R. L. Byers, Cleveland, Ohio.

5:30 P.M. Social Hour (Hotel Statler—Euclid Ballroom)

6:30 P.M. Past Presidents' Dinner (Hotel Statler—Tavern Room)

Wednesday, January 29

8:30 A.M. REGISTRATION (Hotel Statler—Mezzanine Floor)

9:30 A.M. TECHNICAL SESSION (Hotel Statler—Grand Ballroom)
Human Tolerance to Heat by Willard Machle, M.D.
Methods Used in Determining the Health Hazards Arising from the Inhalation of Various Chemicals by F. F. Heyroth, M.D.
Minimal Replenishment Air Required for Living Spaces by W. V. Consolazio and L. J. Pecora.

12:00 NOON Exposition (Lakeside Hall)

12:45 P.M. INSPECTION TRIPS
Laboratory of National Advisory Committee for Aeronautics
Nela Park Lighting Institute and A. G. A. Laboratory

6:30 P.M. Hospitality Hour (Hotel Statler—Parlors 1, 2 and 3 Mezzanine Floor)

7:30 P.M. Annual Banquet (Hotel Statler—Grand Ballroom) Toastmaster: Lester T. Avery
Presentation of Past President's Emblem to Alfred J. Offner by Homer Addams
Speaker: Dr. Kirtley Mather
Subject: Science and the Future
Music by Dick O'Haren's Orchestra

Thursday, January 30

9:30 A.M. TECHNICAL SESSION (Hotel Statler—Euclid Ballroom)
Response and Lag in the Control of Panel Heating Systems by F. W. Hutchinson

Unfinished Business
New Business
Installation of Officers
Adjournment

12:00 NOON Exposition (Lakeside Hall)

2:30 P.M. JOINT SESSION with *National Warm Air Heating and Air Conditioning Association* (Public Auditorium—Ballroom)

Proposed Design Procedure for Large, Mechanical Warm Air Heating Systems by S. Konzo, R. J. Martin, D. S. Levinson and R. W. Roose

Determining and Reducing the Concentration of Air-Borne Micro-Organisms by Matthew Luckiesh and A. H. Taylor

Friday, January 31

12:00 NOON to 6:00 P.M. Exposition (Lakeside Hall)

COMMITTEE ON ARRANGEMENTS

D. L. TAZE, *General Chairman*

Honorary Chairmen

LESTER T. AVERY — C. F. EVELETH — G. L. TUVE

Vice-Chairmen

E. B. CARY — P. D. GAYMAN — F. A. KITCHEN — L. S. RIES

J. M. BLACK, *Secretary*

Banquet — JOHN JAMES, *Chairman;* WALTER BAGGALEY, *Vice-Chairman;* R. W. DICKSON, JR., D. W. GLADIEUX, D. E. MANNEN, E. W. PHILIPS.

Entertainment—R. L. CLARK, *Chairman;* S. R. GUILBERT, E. E. MAURER, L. H. POGALIES, E. J. SABLE, W. B. WATTERSON, R. A. WILSON.

Finance—E. F. MORSE, *Chairman;* H. W. HEISTERKAMP, J. F. PLATZ, J. A. SCHURMAN, A. L. VANDERHOOF.

Inspection—L. G. POWERS, *Chairman,* W. M. ROWE, *Vice-Chairman;* K. A. GOTTSCHALK, L. H. JACKETT, W. P. MILLER.

Laboratory Reception — C. M. HUMPHREYS, *Chairman;* R. M. CONNER, G. V. PARMELEE, J. J. LA SALVIA, R. F. STAMBERGER.

Ladies—R. H. CUTTING, *Chairman;* MRS. R. H. CUTTING, MRS. L. T. AVERY, MRS. P. D. GAYMAN, MRS. C. M. HUMPHREYS, MR. AND MRS. J. P. JONES, MR. AND MRS. W. R. RHOTON, MRS. L. S. RIES, MRS. CYRIL TASKER, MRS. G. L. TUVE.

Publicity — W. R. MOORE, *Chairman;* J. A. DAHLSTROM, K. T. DAVIS, W. E. EYNON, ALEXANDER GALABA, B. L. GAMBLE, EVERETT W. GRAY, R. A. JACK, JOHN RICHMOND, W. R. TELLER.

Reception—J. E. WILHELM, *Chairman;* G. P. NACHMAN, *Vice-Chairman;* J. L. FOLEY, J. L. FRISSE, C. L. GRANDSTAFF, H. L. REPP, W. R. RHOTON, L. E. SLAWSON, H. E. WETZELL, H. K. JENNINGS, (Cincinnati Chapter), R. B. BRENEMAN (Central Ohio Chapter).

Sessions—R. L. BYERS, *Chairman;* E. E. MAURER, *Vice-Chairman;* P. M. BERRY, J. C. BOEHM, JR., L. C. BURKES, H. F. CURTIS, T. D. DRAVAGE, J. R. KINKAID, A. E. LAVELLE, H. A. TOLERTON.

Transportation — R. E. SHERMAN, *Chairman;* L. W. DUNBAR, P. B. FLEMING, J. A. HALL, C. A. MCKEEMAN, C. R. MATTHEWS.

1305

No. 1305

NATURE OF AIR FLOW AT SUCTION OPENINGS

By ALLEN D. BRANDT,* RUSSELL J. STEFFY,** AND RICHARD G. HUEBSCHER,**
CLEVELAND, OHIO

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio, in cooperation with the U. S. Public Health Service.

INTRODUCTION†

THIS paper is the second report of research investigations made by the Society, in cooperation with the *U. S. Public Health Service,* to provide fundamental data for those engaged in the design and construction of industrial exhaust systems. The first paper[1] reported the results of laboratory studies on the energy losses at the entrance to 175 different types of suction openings. Coefficients of entry to the hoods were calculated and the effects of various factors on the coefficient of entry were discussed.

Dr. Brandt was assigned to the A.S.H.V.E. Research Laboratory by the *U. S. Public Health Service,* from October 1945 to July 1946; the Committee on Research provided the technical assistants and equipment.

Other parts of the overall research program are under way at the Laboratory and elsewhere under the guidance of a committee of engineers experienced in the field of industrial atmospheric sanitation. The results will be reported to the membership as the program progresses. Comments and suggestions from members and others will be welcomed. They should be addressed to the A.S.H.V.E. Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio.

PREVIOUS STUDIES

The importance of local exhaust ventilation in industrial atmospheric sanitation has been recognized for many years. It was not until about 1930, however,

---

*Sanitary Engineer, U. S. Public Health Service, assigned to A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.
**Research Assistant, A.S.H.V.E. Research Laboratory.
†Cyril Tasker, Director of Research A.S.H.V.E.
[1]Exponent numerals refer to Bibliography.
Presented at the 53rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January 1947.

that the nature of air flow at suction hoods was studied in an attempt to develop fundamental relationships between the shape, size and type of hood; the velocity *in front of the hood;* and the volume rate of air flow through the hood. Possibly the most comprehensive and the first of such investigations was undertaken by DallaValle,[2] and most of his findings are in common use today by designing engineers. He reported that the relationship between axial velocity in front of freely suspended, unflanged hoods and the quantity ventilation rate through the hoods is expressed with sufficient accuracy for all practical purposes by the equation,[3]

$$\frac{Y}{100 - Y} = 0.1AX^{-2} \qquad (1)$$

where

Y = percent of face velocity found at distance X.
A = face area of hood in square feet.
X = axial distance in feet from hood face to point where velocity is Y.

This equation converted to the form commonly used is, [4, 5, 6]

$$Q = V(10X^2 + A) \qquad (2)$$

where

Q = quantity rate of air exhausted through the hood in cubic feet per minute.
V = velocity in feet per minute at axial distance X, in feet, from face of hood.
X = distance in feet from face of hood to point where velocity is V.
A = area of hood face in square feet.

In this study DallaValle used a special Pitot tube[7] to measure the air velocities. This is a velocity-pressure-type instrument requiring a very sensitive pressure gage and extreme care to measure velocities as low as 120 fpm. The development of the heated-thermometer anemometer, about 1935, resulted in a velocity measuring device which permitted the measurement of much lower velocities with comparative ease.[8] In the early forties, Silverman, using the heated-thermometer anemometer, attempted to extend DallaValle's earlier studies into the lower-velocity range.[9] He, like DallaValle, found that the centerline velocity relationship can be expressed accurately only by complex equations, but that it can be stated with sufficient accuracy for all practical purposes by an equation of the form used by DallaValle, namely:

$$\frac{V}{V_0 - V} = KAX^{-2} \qquad (3)$$

where

V = centerline velocity in feet per minute at distance X, in feet, from the face of the hood.
V_0 = average velocity in feet per minute at the face of the hood.
K = constant.
A = area of hood face in square feet.
X = centerline distance, in feet, from face of hood to point where velocity is V.

For unflanged, freely suspended hoods, Silverman found the value of K to be 0.225, which, when used in Equation 3, may be converted to the more common form as follows:

$$Q = V(4.45\,X^2 + A) \qquad (4)$$

where

Q, V, X and A are the same as in Equation 2.

A glance at Equations 2 and 4 shows that they will give widely different results—over 100 percent difference for many values of X and A. For example, the exhaust rate required to produce a velocity of 100 fpm at a distance of 9 in. from a freely suspended 4-in. diameter duct end is 570 cfm by DallaValle's equation, and 260 cfm by Silverman's. These results pose a serious problem for the engineer who designs local exhaust systems, since he must decide whether to use Equation 4 with the advantageous lower exhaust rate but the possible disastrous result of inadequate control, or to use DallaValle's data and be criticised for the high exhaust rate and heating load in cold weather.

PRESENT STUDY

The present study was undertaken to find out which, if either, of these equations is correct, and to provide the engineer with the factual information needed

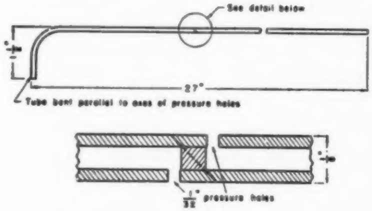


FIG. 1. SPECIAL PITOT TUBE

to design systems which will work effectively. The test arrangement was the same as described in the Bibliography.[1] Velocity measurements in the front of the hoods were made with two types of Pitot tubes and a heated-thermometer anemometer. The three instruments were not used to measure the velocity at each point studied, but were used at sufficient points to determine whether the results obtained by them were comparable. Some of the pressures, when the Pitot tubes were used, were measured with ordinary inclined U-tube manometers, some with a two-liquid differential gage, and some with a commercial type of inclined gage. The one tube, designated the special Pitot tube, was similar in many respects to that used by DallaValle.[7] It was made by cutting a ⅛ in. outside diameter piece of copper tubing at an angle of about 45 deg, sealing the hole in each piece of the tubing, soldering the copper tubes together again, and drilling a hole in each portion of the tube so that the holes were essentially opposite each other and facing in opposite directions (see Fig. 1). The other tube, designated the small Pitot tube, was a scale model of the standard Prandtl instrument and had a tube diameter of only about 1/16 in. at the end with the pressure holes.

The instruments were calibrated in a wooden tunnel as described in the Bibliography.[8] It is interesting to note that the special Pitot tube used in this

study produced a pressure reading 1.78 times the true velocity pressure, a figure which agrees very well with the 1.60 to 1.75 reported by DallaValle for his special Pitot tube. After the study had been completed, the calibration of the instruments was checked in free air, as will be described later.

Since it was the purpose of this investigation to study only the axial or center-line flow relationship, readings were made only on the axial line of the hoods at 2, 3, 4, 5, 6, 9, 12, 15, and 18 in. from the faces of the hoods. The middle of the bulb of the heated thermometer, the middle of the special Pitot tube at the pressure holes, and the static holes of the small Pitot tube were located at the point in question as carefully as possible without the use of special mechanical devices for centering them. The small Pitot tube was employed just as if an ordinary velocity pressure reading were being made, but the static holes were located at the point where the velocity was to be measured, and only the pressure produced by the static holes was measured.

UNFLANGED HOODS

A total of 9 unflanged hoods was studied. Details on the size and shape of these hoods are given in Table 1. Six round, one square, and two rectangular hoods comprise the group. Duct ends and flared hoods were also investigated.

The data obtained for each hood were plotted on log-log paper as shown, for example, in Figs. 2, 3, and 4, and the equation in the form

$$\frac{V}{V_0 - V} = \frac{KA}{X^n} \qquad (5)$$

was then derived from the *curve of best fit*. The values of K and n for each hood are listed in Table 2, in which the hood numbers are the same as in Table 1. It is apparent from the table that the values for round openings are of the same order of magnitude as for rectangular and square hoods and the results are therefore applicable to all the usual shapes of unflanged exhaust hoods. The data given in Table 2 indicate that the axial flow relationship may be expressed by the equation

$$Q = V(9.35X^{2.03} + A) \qquad (6)$$

TABLE 1—UNFLANGED HOODS STUDIED

HOOD No.	SHAPE	THROAT DIA. IN.	FACE DIMENSIONS IN.	INCLUDED ANGLE (DEG)
1	Round	8	8 dia.	0
2	Round	4	9 dia.	40
3	Round	8	11.3 dia.	50
4	Round	8	18 dia.	58
5	Round	12	27 dia.	40
6	Round	4	4 dia.	0
7	Square	8	16 x 16	45
8	Rect.	8	11 x 22	54 and 12½
9	Rect.	8	8 x 31½	107 and 0

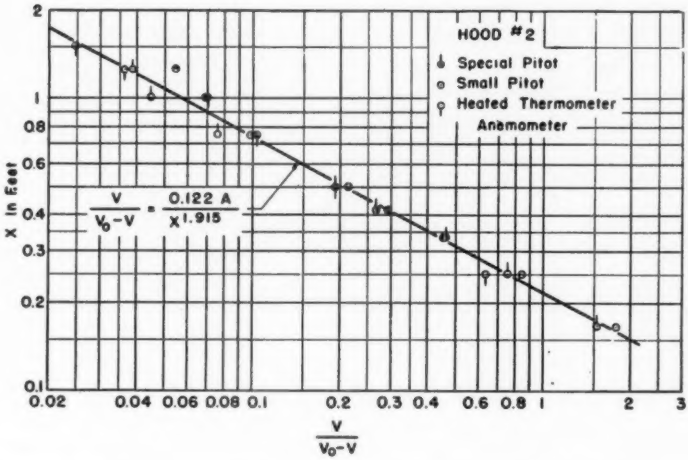


FIG. 2. CENTERLINE FLOW RELATIONSHIP FOR HOOD NO. 2

This equation checks rather well with that derived by DallaValle, and it therefore seems reasonable to conclude that the convenient equation

$$Q = V(10X^2 + A)$$

which is in common use today, may continue to be used by designing engineers for unflanged, freely suspended round, square, or rectangular exhaust hoods of the usual types.

Before proceeding with the other parts of this investigation, it may be well to consider the physics of air flow into suction openings, to see whether the foregoing relationship appears reasonable. It is obvious, of course, that if a point source of suction could be created, the air would flow into that point equally from

Table 2—Values of K and n in Equation $\frac{V}{V_0 - V} = \frac{KA}{x^n}$ for Unflanged Hoods

HOOD NO.	K	n
1	0.0854	2.230
2	0.1220	1.915
3	0.1090	2.000
4	0.1170	2.000
5	0.1300	1.950
6	0.0950	2.110
7	0.1260	2.280
8	0.0876	2.000
9	0.0915	1.810
Average	0.1071	2.033

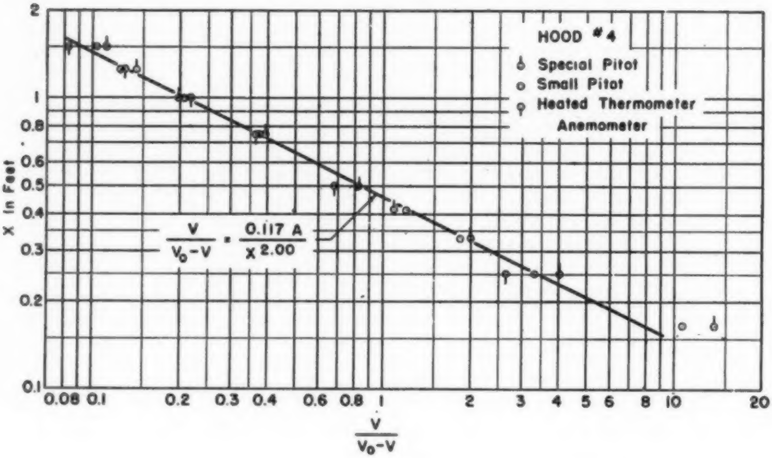


FIG. 3. CENTERLINE FLOW RELATIONSHIP FOR HOOD NO. 4

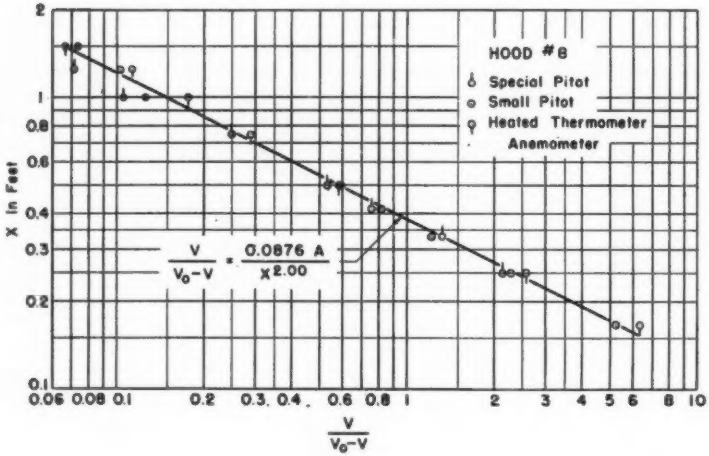


FIG. 4. CENTERLINE FLOW RELATIONSHIP FOR HOOD NO. 8

all directions, and the velocity contours would be spherical surfaces. Since the area of the surface of a sphere is $4\pi r,^2$ the air flow relationship would be:

$$Q = 4\pi r^2 V = 12.57 r^2 V \quad (7)$$

where

Q = the quantity rate of air flowing into the point source of suction in cubic feet per minute.

r = radius in feet of the spherical surface at any point in which the air velocity is V feet per minute.

V = the velocity in feet per minute at any point in the spherical surface r distance in feet from the point source of suction.

When air is moving into a hood or pipe opening under suction, these theoretical conditions do not prevail, but it is reasonable to assume that with a small-diameter pipe end, for instance, the velocity contours would be almost spherical in shape and would have their centers approximately at the point formed by the intersection of the axis of the pipe with the plane of the face of the opening. To make Equation 7 applicable to all conditions, even if r becomes zero, as would be the case if the velocity were measured in the face of the hood, a correction factor must be inserted in the area portion ($12.57r^2$) of the equation. This condition is met if A, representing the face area of the hood, is included in the existing area factor, because the equation then becomes $Q = AV$ when r is zero. Equation 7 now becomes:

$$Q = V(Kr^2 + A)$$

in which K is somewhat less than 12.57.

Since X of Equations 2 and 4 is the same as r of Equation 7, it becomes:

$$Q = V(KX^2 + A) \quad (8)$$

where

Q, V, X and A are the same as in Equation 2.

K = some value less than 12.57.

Since the contours are not truly spherical surfaces, as was the case with the theoretical conditions, the constant as well as exponent of X will probably change, but it seems likely that Equation 8 should represent fairly closely the true relationship. For the constant in this equation to decrease substantially, it is necessary that the contours in front of suction openings elongate considerably in an axial direction. Since this seems unlikely, it appears that Equation 2 should be more nearly correct than Equation 4.

FLANGED HOODS

Three flanged hoods (one with three different size flanges) were also investigated. Details on these hoods and flanges are given in Table 3. Airflow relationship data were plotted on log-log paper similar to that for unflanged hoods. A straight line was found to fit the points for two hoods, Nos. 11 and 12, but not for the other three. The points obtained for hoods Nos. 13 and 14 defined curves (see Figs. 5 and 6) while those obtained for hood No. 10 appear to define

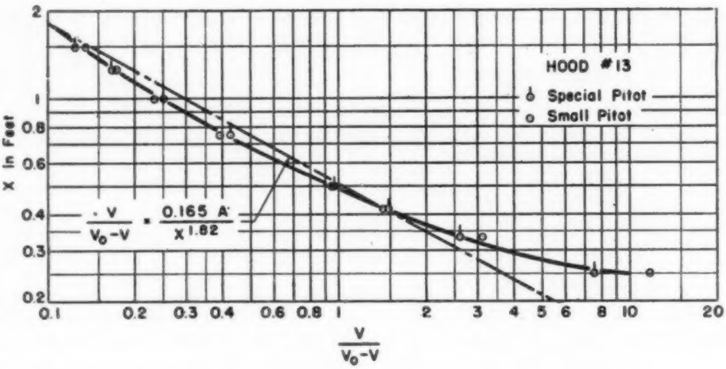


FIG. 5. CENTERLINE FLOW RELATIONSHIP FOR HOOD NO. 13

TABLE 3. FLANGED HOODS STUDIED

HOOD NO.	SHAPE	THROAT DIA. IN.	FACE DIMENSIONS IN.	INCLUDED ANGLE (DEG)	FLANGE SIZE IN.	DIST. HOOD EDGE TO FLANGE EDGE IN.
10	Round	8	8 dia.	0	24 x 24	8
11	Round	8	18 dia.	58	48 x 48	15
12	Square	8	16 x 16	45	48 x 48	16
13	Square	8	16 x 16	45	24 x 24	4
14	Square	8	16 x 16	45	32 x 32	8

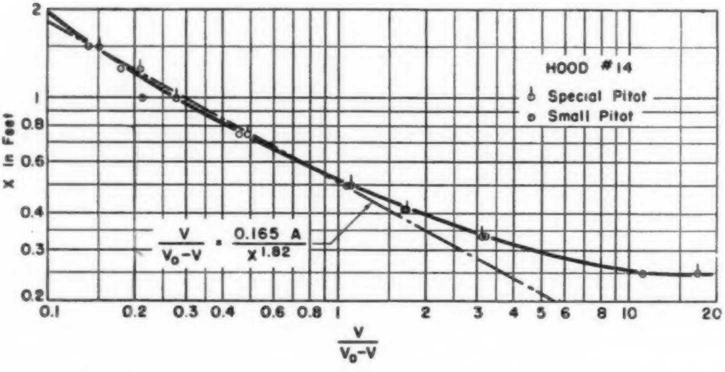


FIG. 6. CENTERLINE FLOW RELATIONSHIP FOR HOOD NO. 14

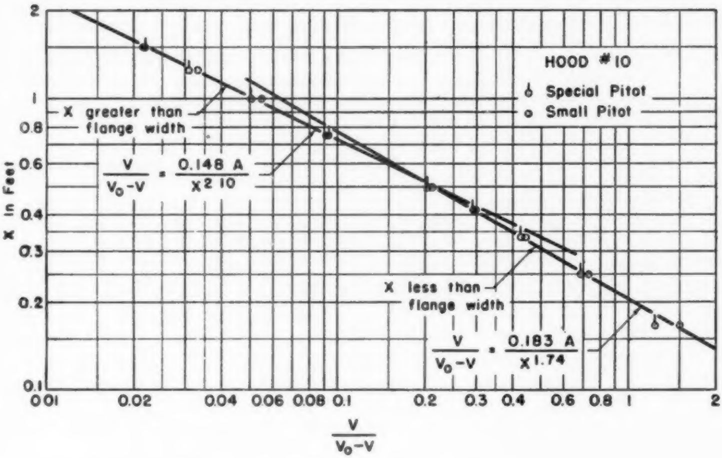


Fig. 7. Centerline Flow Relationship for Hood No. 10

two different straight lines (see Fig. 7). While the data are too few to permit conclusions, they suggest that a straight line relationship will exist with flanged hoods if the flange width is not less than the distance from the face of the hood to the point where the velocity is measured, but that such relationship will not exist if the flange width is small as compared to the distance X.

The data suggest also that the air flow relationship for flanged hoods is expressed by the following equation, so long as X is not larger than the flange width:

$$Q = V(6.06X^{1.82} + A) \quad \text{.......................(9)}$$

Even though the results obtained for hoods 10, 13, and 14 do not define straight line relationships, the use of Equation 9 will obtain results sufficiently accurate for practical purposes for all flanged hoods (see Table 4).

Since Equation 9 is not particularly convenient for everyday use, it may be desirable for designing engineers to use the equation for an unflanged hood (No. 11) and to make appropriate corrections for flanges. Thus the value of Q as determined by Equation 2 may be reduced as much as 35 percent if the hood is equipped with flanges at least as wide as the distance X in the equation. For narrower flanges the percentage reduction should be decreased proportionately.

A rectangular unflanged hood having face dimensions of 8 in. x 31½ in. was investigated while it was lying on a large table top to simulate conditions in industry when hoods are used at work benches. The hood was No. 9 and had one of the long sides in contact with the table top. Since this hood was studied also as a freely suspended unflanged hood, the results may be compared, to

TABLE 4. COMPARISON OF MEASURED VELOCITIES AT FLANGED HOODS WITH THOSE COMPUTED BY $Q = V(6.06x^{1.82} + A)$

Hood No.	Velocity at Different Distances from Hood Face															
	3 In.		4 In.		5 In.		6 In.		9 In.		12 In.		15 In.		18 In.	
	Meas	Comp[a]	Meas	Comp[a]	Meas	Comp[a]	Meas	Comp[a]	Meas	Comp[a]	Meas	Comp[a]	Meas	Comp[a]	Meas	Comp[a]
10	2905	2970	2133	2125	1591	1570	1211	1200	597	625	352	381	218	257	150	187
11	1139	1200	956	1050	817	906	703	780	467	505	316	343	212	246	173	185
12	1396	1175	1142	1020	943	885	802	762	494	494	325	336	218	241	161	182
13	1354	1185	1110	1030	887	893	732	768	435	498	293	339	216	243	172	183
14	1355	1150	1106	1000	917	866	758	745	467	483	285	329	236	235	183	178
Avg. dev. (%)		10.2		7.5		4.9		3.7		6.1		10.3		12.4		10.7

[a]Computed from equation given at top of table.

TABLE 5. INFLUENCE OF LARGE FLAT SURFACE ADJACENT TO LONG SIDE OF RECTANGULAR HOOD

Distance from Hood Face in In.	Centerline Velocity in Feet per Minute	
	Hood Freely Suspended	Hood on Table
2	1353	1511
3	1081	1185
4	798	990
5	641	862
6	520	695
9	332	507
12	163	364
15	123	275
18	89	212
Equation of flow	$Q = V(10.9\ x^{1.81} + A)$	$Q = V(5.3\ x^{1.87} + A)$

TABLE 6. COMPARISON OF RESULTS OBTAINED WITH SPECIAL PITOT (SP.P), SMALL STANDARD PITOT (SSP) AND HEATED-THERMOMETER ANEMOMETER (HTA)

	Less than 300 fpm (Avg of 21 Readings)			300 to 600 fpm (Avg of 18 Readings)			Over 600 fpm (Avg of 15 Readings)		
	Sp.P	SSP	HTA	Sp.P	SSP	HTA	Sp.P	SSP	HTA
Avg. velocity in fpm	170	189	212	434	441	468	1179	1215	1123
Sp.P coefficient based on SSP results	1.44			1.71			1.68		
Percent of HTA from SSP			+12.2			+6.1			−7.6

demonstrate the improvement produced by the adjacent table top (see Table 5). It is interesting to observe the different equations resulting from these studies. For the freely suspended unflanged hood the centerline flow relationship was found to be:

$$Q = V(10.9X^{1.81} + A),$$

and when the hood was adjacent to the table top the relationship was:

$$Q = V(5.3X^{1.67} + A).$$

Comparison of Velocity Measurements

Even though no conspicuous and consistent discrepancy was apparent between the readings obtained by the different instruments while the data were being collected, during analysis of the data it appeared that the velocities read with the heated-thermometer anemometer were higher than those read with the two Pitot tubes in the low-velocity range. To study this observation further a significant number (54) of points, at which the velocity had been measured with the three instruments, were selected at random and divided into convenient velocity ranges which would give fairly even distribution of the readings among the groups. The recorded velocities in each group were then averaged for each instrument. These results are summarized briefly in Table 6, from which it can be seen readily that, if it is assumed that the true velocity is that recorded by the small standard Pitot tube, the coefficient of the special Pitot tube is lower in free air (average 1.61) than in a tunnel (average 1.78), and the heated-thermometer anemometer in free air gives high results in the low-velocity range and low results in the high range. The percentage of error increases as the velocity range decreases.

The lower coefficient of the special Pitot tube may be due in part also to the lower average velocity encountered in free air. This agrees with the findings of Ingram[10] and others that, in unconfined air near suction openings, pressure tubes (special Pitot tubes) read progressively lower as the velocity decreases, and progressively higher as the velocity increases, when compared with the heated-thermometer anemometer.[10] Apparently, on the assumption that the true velocity was given by the heated-thermometer anemometer, these authors attributed this phenomenon to changing coefficient for the tubes tested.

It was realized, at the initiation of these tests, that there might be discrepancies between the results obtained from the readings of the several instruments used. However, since the tests were being made to investigate variations of the order of 100 percent, extremely precise instrumentation was not considered to be essential.

To serve as a double check on the findings, particularly in the low-velocity range, the instruments were recalibrated in free air. A new and perhaps useful, air-flow arrangement was devised for this purpose.

Calibration in Free Air

To approximate the conditions that would obtain if a point source of suction were possible, a hollow copper sphere, 6⅞₁₆ in. in diameter, was drilled full of ¼ in. holes. A 4-in. diameter hole was then cut in the copper ball and it was

attached to a 4-in. diameter pipe. A collar, in the shape of a frustum of a cone, was then fitted around the pipe where the copper sphere was attached, in such fashion that it formed the extension of the cone produced by the imaginary surface formed by drawing lines from the center of the sphere to the circumference of the 4-in. circle where the pipe and ball joined (see Fig. 8). Since the resistance to air flow through the holes in the copper sphere was considerable, this ball became in effect a very large point source of suction, except for the portion attached to the pipe. When air was exhausted through the copper sphere, uniform air flow toward the ball was created in the entire spherical zone beyond the ball, with the exception of the portion cut off by the collar. Obviously, then, if extraneous air currents could be eliminated, the velocity contours would be portions of spherical surfaces and the true velocity at any distance from the copper sphere could be computed accurately for any volume rate of air flow.

As preliminary tests indicated that it was practically impossible to eliminate extraneous air currents entirely, readings were taken at five different stations

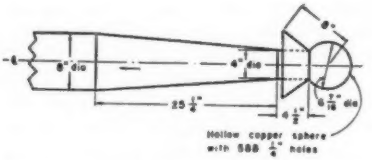


FIG. 8. TEST DEVICE FOR CHECKING INSTRUMENT CALIBRATION

around the sphere and averaged to give the measured velocity at that distance from the copper ball. These stations were above and below the ball in its vertical axis, at each side of the ball in the horizontal axis parallel to the face of the 4-in. pipe, and in front of the ball in the centerline of the 4-in. pipe. Velocity measurements were made at 2 in., 3 in., 4 in., and 6 in. from the copper sphere with the special Pitot and the small Pitot, and a few readings were taken at 4 in. and 6 in. with the heated-thermometer anemometer. As the velocity at 6 in. was too low to give trustworthy results with the small Pitot tubes, these results were discarded. The data obtained with the two Pitot tubes are summarized in Table 7, from which it is apparent that the small Pitot tube gives remarkably accurate results in free air. The greatest error in the velocity measured with this instrument appears to be only 2.8 percent and the average error for the three distances is only 1.8 percent. The coefficients of the special Pitot tube for the four different distances are 1.66, 1.64, 1.61, and 1.30. These values, in conjunction with those given in Table 6, indicate that the coefficient of the special Pitot in free air decreases as the velocity decreases. Only a few readings were obtained with the heated-thermometer anemometer, and conclusions therefore are not warranted. The few measurements which were made indicated that this device was reading in the order of 20 percent high at 4 in. and 6 in. from the copper sphere.

TABLE 7. CALIBRATION CHECK OF TWO PITOT TUBES IN FREE AIR

DIST. FROM SPHERE IN.	COM-PUTED	Velocity in Feet per Minute											
		Special Pitot						Small Standard Pitot					
		Reading Station					Avg.	Reading Station					Avg.
		1	2	3	4	5		1	2	3	4	5	
2	553	474	560	533	549	552	534	489	585	557	578	585	558
3	389	333	387	354	392	398	373	365	411	371	419	415	396
4	253	234	246	237	237	246	240	238	242	242	247	259	246
6	155	143	151	117	127	122	132	—	—	—	—	—	—

CONCLUSIONS

The results of this investigation appear to substantiate DallaValle's findings and suggest an explanation for the widely different results reported by Silverman. The centerline flow relationship at unflanged suction openings is expressed with sufficient accuracy for practical purposes by the convenient equation $Q = V(10X^2 + A)$. No similar, convenient, and fairly accurate equation is applicable for flanged openings. The equation $Q = V(6X^{1.8} + A)$ may be used for flanged hoods, but it is more convenient to use the unflanged relationship and to correct the results for flanging.

BIBLIOGRAPHY

1. A.S.H.V.E. RESEARCH REPORT No. 1295—Energy Losses at Suction Hoods, by A. D. Brandt and R. J. Steffy. (A.S.H.V.E. TRANSACTIONS, Vol. 52, 1946, p. 205.)

2. Studies in the Design of Local Exhaust Hoods, by J. M. DallaValle. (Thesis for degree of Sc. D., Harvard Engineering School, 1930.)

3. Studies in the Design of Local Exhaust Hoods, by J. M. DallaValle and Theodore Hatch. (*A.S.M.E. Transactions,* Vol. 54, 1932, p. 31.)

4. Industrial Exhaust Ventilation in Industrial Hygiene, by A. D. Brandt. (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 331.)

5. A Summary of Design Data for Exhaust Systems, by A. D. Brandt. (*Heating and Ventilating,* May 1945, Vol. 42, p. 73.)

6. Industrial Dust, by Philip Drinker and Theodore Hatch. (McGraw-Hill Book Co., New York, 1936.)

7. The Determination and Control of Industrial Dust, by J. J. Bloomfield and J. M. DallaValle. (*U. S. Public Health Bulletin No. 217,* Government Printing Office, Washington, D. C., 1935.)

8. The Heated-Thermometer Anemometer, by C. P. Yaglou. (*Journal Industrial Hygiene and Toxicology,* 1938, Vol. 20, p. 497.)

9. Centerline Velocity Characteristics of Round Openings Under Suction, by L. Silverman. (*Journal Industrial Hygiene and Toxicology,* 1942, Vol. 24, p. 259.)

10. The Characteristics of Double Pitot Tubes, by F. R. Ingram, E. Diez-Canseco and L. Silverman. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning,* November 1942, Vol. 14, p. 702.)

APPENDIX

Shortly after the foregoing article was completed, a report of a similar study made on one hood of much larger size came to the authors' attention. This report, prepared by Knowlton J. Caplan, gives the results of a post-graduate research project which he carried out at Michigan State College during the spring and summer term of 1946. The hood he studied was 8 ft. square, with one edge on the floor and 15-in. wide flanges at the other three sides.

It is not the intent of this Appendix to report or discuss the findings of this investigation, but merely to cite two interesting trends in the results which parallel trends in the author's data—trends which were either not mentioned or not stressed owing to paucity of data. These tendencies will now be mentioned for what they are worth. More research is needed to substantiate or refute these findings. The following data are included in this paper with the permission of Prof. L. G. Miller, Michigan State College, and Mr. Caplan, Michigan Department of Health.

Caplan also used three velocity measuring instruments, but the only device common to both studies was the heated-thermometer anemometer. His other two instruments were the Alnor Velometer and a rotating vane anemometer. The calibration of the instruments was not checked by Caplan; the manufacturer's calibrations were used

TABLE 8. COMPARISON OF RESULTS OBTAINED WITH THE VELOMETER (*V*), THE ROTATING VANE ANEMOMETER (*RVA*) AND THE HEATED-THERMOMETER ANEMOMETER (*HTA*)

	100 TO 150 FPM (AVG OF 8 RDGS)			150 TO 200 FPM (AVG OF 6 RDGS)			200 TO 300 FPM (AVG OF 3 RDGS)			300 TO 400 FPM (AVG OF 2 RDGS)		
	V	*RVA*	*HTA*	*V*	*RVA*	*HTA*	*V*	*RVA*	*HTA*	*V*	*RVA*	*HTA*
Avg vel in fpm	118	99	138	169	153	178	263	238	253	385	324	312
Deviation of *HTA* from *RVA*			+39%			+16%			+6%			−4%
Deviation of *HTA* from *V*			+17%			+5%			−4%			−19%

throughout. In Table 8 are summarized the air velocity determinations at all points where readings were taken with the three instruments—all centerline readings. For convenient comparison the results in Table 8 have been arranged somewhat similar to the results in Table 6. It is apparent from Table 8 that while the magnitude of the deviations of the heated-thermometer anemometer values from those given by the other instruments is not the same as shown in Table 6 the trend is unmistakable.

If these trends are correct, Silverman's findings as given in Bibliography[9] might likely be due to high velocity values obtained with the heated-thermometer anemometer since most of his reported velocity readings were less than 100 fpm; a range where the true velocity might well have been only about one-half as high as reported. As stated previously, more data are needed to establish or invalidate this suggested trend.

The other similarity noticed in the results of these two reports is the influence of flanges on the centerline velocity relationship to the quantity rate of exhaust through the hood. From Caplan's readings, plotted as shown in Figs. 2 to 7, the points within the zone of influence of the flanges appear to define a straight line, and beyond this zone of influence define a different straight line (Fig. 7), the breakpoint being at a distance from the hood which is greater than the flange width but smaller than the distance from the center of the hood face to the edge of the flange. Whether this suggested characteristic is correct, and what relationship there is between the breakpoint and the flange, or flange plus hood width, cannot be determined from the few results reported in Caplan's report and the authors' paper. More research is needed also on this point.

DISCUSSION

LESLIE SILVERMAN,†† Boston, Mass. (WRITTEN): The writer is very much interested in this article since it involves certain data which were obtained in this laboratory several years ago.

In fitting our data, we favored a dimensionless equation of the form

$$\frac{V}{V_0} = K\left(\frac{X}{\sqrt{A}}\right)^n$$

since this problem is essentially one of fluid dynamics. It will be noted that we cited limits of $\frac{X}{\sqrt{A}}$ between 0.5 and 3.0 for our measurements. In the application of DallaValle's function to our data, the values of the exponent on X varied from 1.4 to 2.5, so that any equation fitted was obviously very approximate—Dr. Brandt's work confirms this fact.

A study of the data presented in Table 2 by Brandt, Steffy and Huebscher shows the approximation of their data. Thus, in Table 2, if we take the values of K for a constant exponent $n = 2.00$ we get:

Hood Number	K	n	Deviation from DallaValle's constant $K = 0.1$ Percent
3	0.1090	2.00	+ 9.0
4	0.1170	2.00	+17.0
8	0.0876	2.00	—12.4

Hood 5 may be included also since its exponent is very close (within 2½ percent of 2), or

5	0.1300	1.950	+30.0[a]

[a]This value assumes $n = 2$ for hood 5 with the value of K cited.

In the foregoing values, if DallaValle's constant of $K = 0.1$ is employed for reference it can be seen that Brandt's deviation varies from —12.4 percent to +30 percent. In view of these wide differences it seems unnecessary to express values of K and n to 4 places, as in Table 2 of this article, especially since the curves are determined *by eye* according to best fit and not by use of the method of least squares. Further, the points plotted in Figs. 2 to 7 show differences which would only warrant n being expressed to the nearest tenth and K to the nearest hundredth.

In using the DallaValle type of equation, two important facts must be stated which we emphasized in our paper. When the distance X from the hood is zero, Equation 2 reduces to $Q = AV$, where V is the centerline velocity. In only frictionless air, flowing into a perfect hood or bell-mouth opening with a flat contour and no turbulence on entrance, would this be correct? The actual centerline velocity at the mouth as shown by our data, may be as high as twice the average face velocity V_0, since an unflanged plain opening does not flow full. In our work we had difficulty in reproducing velocity measurements close to the mouth, because of the unstable turbulent conditions. It is evident that the centerline velocity at the mouth will exert some influence on the velocity along the centerline. Thus, in our Fig. 2 (see Bibliography[8]), it can be seen that for a 20 in. pipe opening the point velocity is not reduced to that of Q/A until 5 in. from the opening.

Because of the vortex at the pipe mouth, it is obvious that the equation proposed by DallaValle gives erroneous results close to the opening.

The second fact which must be borne in mind is that at infinite distance ($X = \infty$) the velocity is not zero, as given by this type of equation, but is equal to the extraneous

††Assistant Professor of Industrial Hygiene, Harvard School of Public Health.

air currents, since these are always present in the atmosphere of industrial operations. We feel that the latter condition may have influenced our results, since measurements were made at distances as great as 40 in. from the opening. At this distance the actual velocities measured were only slightly above room air currents. The heated thermometer which was employed is an integrating instrument, and therefore is influenced by the turbulence or any factor which contributes to heat loss from the heated thermometer bulb.

In other studies, Brandt's test results with the thermo-anemometer agreed reasonably well with ours. It would appear from these data that the differences must be instrumental, as he suggests. In an article* published previously he presents data on a slot type hood with the heated thermometer anemometer. If one measures his velocity contours and calculates the quantity flowing, the results obtained are:

Actual quantity exhausted, cfm	1320
Measured value from plotted contour, 300 fpm	1650
Measured value from plotted contour, 200 fpm	1650
Measured value from plotted contour, 150 fpm	1650
Measured value from plotted contour, 100 fpm	1740

These values, then, indicate quantities determined from velocity readings to be 25 to 32 percent higher than the actual metered pipe flow. A similar study‡ of data reported on the same type of hood, by another laboratory using the heated thermometer, shows results indicating that contour values are as follows:

Actual quantity exhausted, cfm	1000
Measured value from plotted contour, 200 fpm	1300
Measured value from plotted contour, 100 fpm	1475

These values indicate a variation of from 30 to 47.5 percent above the actual exhausted quantity.

Three different laboratories (four if Caplan's results are included) thus have obtained high values independently when using the heated thermometer anemometer (our values for hoods fall into this category since we obtained deviations of approximately +40 percent as discussed earlier). It would appear, then, that the heated thermometer anemometer deserves further investigation. In this study, Brandt shows heated thermometer readings which average as high as 20 percent above other measuring instruments using his calibration method. Therefore the writer would like to comment on these discrepancies.

The heated thermometer anemometer is an integrating instrument and measures essentially heat loss created by air movement. This air movement may be due to turbulence, as well as linear motion of air. Pitot tubes measure point velocity, and in front of openings only static pressure at a point is measured. To date, heated thermometer anemometers have been calibrated in wind tunnels and good agreement has been obtained on duplication of these calibrations by other observers.¶ No one has attempted to compare calibrations made outside an opening under suction where the flow is presumed to be streamline, nor has it been possible to calibrate the thermo-anemometer with rotating arm devices (thermocouples might be used for this purpose). Fundamentally, then, the instrument should be calibrated under conditions similar to those actually used. The heated thermometer is also susceptible to stem immersion correction. In Brandt's Table 6 he compares random selected values and indicates high readings at less than 300 fpm, and 300 to 600 fpm, and low readings above 600 fpm. Since there is no mention of it the writer wonders whether any attempt was made to correct for stem immersion on any of these readings. Yaglou (see Bibliography[8]) shows that the stem immersion correction may be as high as 30 percent: that is,

*Ventilation of Plating Tanks, by Allen D. Brandt. (*Heating, Piping & Air Conditioning*, July 1941, p. 434.)

‡New Data for Practical Design of Ventilation for Electroplating, by William P. Battista, Theodore Hatch and Leonard Greenburg. (*Heating, Piping & Air Conditioning*, February 1941, p. 81 and June 1941, p. 365.)

¶A.S.H.V.E. RESEARCH REPORT No. 1140—The Use of Air-Velocity Meters, by G. L. Tuve, D. K. Wright, Jr. and L. J. Seigel. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 645.)

uncorrected readings are from 0 to 30 percent lower than actual, depending upon the indicated velocity and depth of immersion. Corrections are necessary for all velocities above 100 fpm. Thus, in Brandt's Table 6 some values may be influenced by immersion. Only at large openings or substantial distances where the contour is flat can the stem be subject to the same velocity as the bulb. Therefore, in these authors' sphere, calibration procedure, the actual deviation of the heated thermometer may be even greater than +20 percent, and the values in Table 6, last column, may be positive rather than negative. In our work we tried to correct for immersion by several means, as reported in Bibliography[9].

It thus appears that the differences in velocities may be due to the fact that the heated thermometer is non-directional, whereas point velocity devices are directional.

Before concluding, the writer would like to comment on other phases of this paper. In his comments on the sphere concept, Brandt considers his reflections on the *physics* of air flow into openings. In our opinion, the discussion which follows is *geometry* not physics. This concept disregards viscosity and density, assumes a perfect fluid, and also neglects turbulence and vortices at the opening. The vortices at the pipe entrance may elongate the contours near the opening, because the centerline velocity is accelerated by the restricted opening produced by vortical flow. This may not be of any importance at substantial distances from the inlet. The sphere theory is a simple concept and valuable for explanation purposes to laymen, but it does disregard the existing fluid dynamics near the entrance.

The sphere method of calibration proposed in this paper may be open to some criticism. It assumes, as in the sphere discussion by Brandt, that calibrations are made with a perfect fluid, which is not the case. If this simple technic is feasible, why bother with rotating arms and similar devices for calibrating instruments in free air? Also, if such a calibration procedure is acceptable for a standard, why go to the extreme of making a sphere? Is not a long narrow slot with cylindrical contours more desirable, and simpler to construct? A slot also has the advantage that measuring instruments are completely immersed, as is necessary in the case of the heated thermometer anemometer, and that the velocity varies directly with distance, making positioning of instruments less critical.

In conclusion, the writer would like to state that he is reconciled to the use of DallaValle's equation, with due regard for its limitations, and since it does refer to point measurements. From the practical standpoint, its use allows adequate safety precaution for contingencies. Because of obvious approximations in fitting measured data and some discrepancy in double Pitot tube coefficients, its accuracy is not very good. I believe that further work on the heated thermometer anemometer for measurements in front of openings is again indicated by this study.

RALPH POOLE, London, England, (WRITTEN): I find that in research papers dealing with air flow there is a general tendency toward difficulty when dealing with low air flow measurements.

In this paper, which gives information that is of real value to practicing engineers, we again find, toward the end of the paper, that we are back in what, I might say, is the mire of low velocity air flow measurement.

The development of the standard Pitot tube, much of which was carried out in the United States by Rouse, has shown us that there are certain inherent errors in the tube, particularly as regards the measurement of the static pressure. This measurement is perhaps the most difficult of all air flow measurements.

The standard Pitot tube is accurate in the measurement of velocity, only because the stem of the tube builds up a positive pressure that corrects the negative pressure error at the static hole. Unfortunately, this self-correction does not occur at air speeds of the order of 200 fpm or less, where error becomes proportionally great. The error is also great when the tube is inclined to the direction of air flow.

I think that, in the tests described by the authors, it is extremely difficult to be certain that the tubes were in fact set in line with the air stream. For such work, we have usually made use of a modified form of the Fechheimer tube, which determines the direction first and insures that the total head and static openings are correctly positioned in relation to the air flow.

We have made extensive use of this tube, which was developed in the United States, and on my first visit here I was quite amazed to find that it is hardly known among the heating and ventilating engineers.

It was developed by Fechheimer, Westinghouse Electric Manufacturing Co., for carrying on investigations of air flow inside electrical machines, in which it was very important to determine both the direction of flow and the static pressure.

Some years ago, we modified the tube still further by making center tapping between the two side tappings to give velocity head. I think I might make that clear with a diagram, Fig. A, which shows a cylinder. The flow is tangential to the cylinder at some angle, approximately 40 deg to the cylinder, so the pressures are measured at those two points and balanced one against the other; either one gives the static pressure. The facing hole, in the front of the cylinder gives the total head. So we have a tube which will actually give total head, static head and direction, as shown in Fig. B.

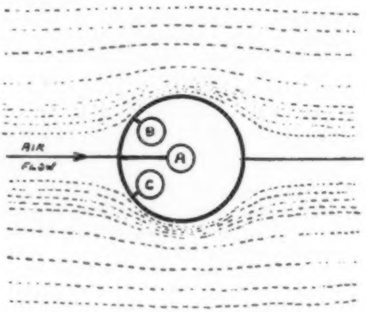


FIG. A. THE MODIFIED FECHHEIMER TUBE

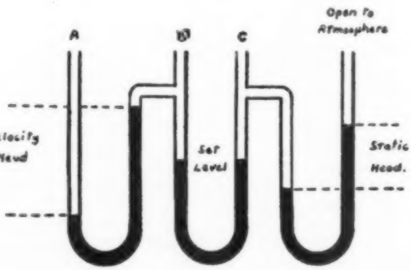


FIG. B. MEASUREMENT OF VELOCITY HEAD AND STATIC HEAD WITH MODIFIED FECHHEIMER TUBE

I say quite sincerely I feel that you owe it, to some extent to Fechheimer, and his colleagues to make more use of the really fine pressure measuring tube which he developed. The other point I would like to make is that all pressure-measuring tubes require a standard. If we dig back, it is very difficult indeed to find what the original standard was for the original Pitot. For low velocity measurements, the calibrations have often been carried out on a whirling arm, so that the tube is whirling around and carries around with it some of the air, and we are left with an error which is indeterminable.

Because of these difficulties, we have developed, in our low velocity wind tunnel, a method which may ultimately become a standard for low velocity calibration purposes. It is a closed circuit wind tunnel with a vertical center section, and we have made special use of a hot-wire anemometer. Having in mind the fact that the wire is susceptible to dust particles or combustible vapors and also that its calibrations are very unstable, we move the wire at a constant speed in the direction of the air flow and measure only its temperature. Nor are we not interested in the precise tempera-

ture. When the wire moves slower than the air, it is cooled; and it is similarly cooled when it moves faster than the air. Therefore, we plot a curve of velocity against temperature, and then determine the speed of movement to the maximum temperature which occurs when the wire is moving at the same speed as the air.

This method appears to give a result as close to absolute velocity as we could obtain. A correction may be made for convection currents set up by the heated wire by merely running the test by two or three different currents. By this means we are able to determine the error that is introduced by convection current.

I would like to have the opportunity to check up the tubes that Dr. Brandt and his colleagues have used on this work, and maybe we shall have the opportunity in the future of sharing the results we get in this special standardizing wind tunnel.

K. J. Caplan, Lansing, Mich.: In the work that was done with that large hood, there would be very little consideration of immersion correction of the heated thermometer, because the hood face was 8 ft across, and your whole thermometer is in the same velocity air stream.

W· N. Witheridge, Detroit, Mich. (Written): We should keep in mind the fact that this study was undertaken principally to settle the status of the commonly quoted DallaValle Equation for centerline flow. My own conclusion is that the present report gives ample evidence that DallaValle's equation is still satisfactory for centerline relationships of airflow into a suction opening, within the practicable range of distances from such openings.

The discrepancies in the literature on this subject seem to be the result of a combination of differences in mathematical treatment, instrumentation, experimental test conditions, and environmental disturbances. There does not seem to be enough data, either in the present paper or in Silverman's report of 1942 to uncover any real errors of manipulation. There are likewise few data on background air disturbances that would clearly indicate that the two laboratory environments are not comparable, but I personally suspect that some of the discrepancy is due to such causes. I am rather favorably impressed by the relatively high air velocities that were used in the current study at Cleveland. I am also glad that a number of different instruments were used in this study, even though the characteristics of each seem to be open to some question.

Air Volume and Velocity Selection

I do not believe that we shall solve this problem until the several laboratories that study air movement toward suction openings arrange to handle enough air through these openings. The exhaust volumes should be high enough so that air velocities at some distance from the hoods will be able to submerge room air disturbances and minimize the errors of instrumentation which occur near the lower limits of sensitivity.

Some laboratories for theoretical studies of this kind have been too small, considering the disturbances created in the room by the operation of the equipment itself, as well as those caused by extraneous and unpredictable causes. A large empty warehouse with a test stand located out in the middle, at a suitable height from the floor, would seem to be more appropriate than one located in some of the cramped quarters used in the past, notwithstanding the trouble that might be encountered in engaging the use of an empty warehouse.

As previous investigators have verified the fact that the positions of the velocity contours are essentially independent of the actual velocity ranges, I believe it would be quite helpful to seek agreement between laboratories using airflows much higher than have so far been studied. It might then be easier to discover whether apparent discrepancies are due to inaccurate or inappropriate instruments, fundamentally different mathematical expressions, or are due to the superimposed effects of cross drafts, thermal disturbances from floor to ceiling, transitions from streamline to turbulent flow, or any other influence that is likely to create dissimilar experimental conditions hitherto not thoroughly reported in the research papers.

I do not favor the dependence upon airflow values below 100 fpm in the development of fundamental data. Regardless of the possibility of instrumental accuracy, the natural test-room air disturbances are too uncertain to be permitted to influence the measurements in space. Only if the fundamental constants had been shown to vary

significantly with hood-face velocities would it be necessary to collect data from a low-velocity experiment.

It might be argued that the reason for conducting research on hood characteristics at values below 100 fpm is that control can be obtained in industry at lower values. This is not true for small hoods, even though some designers have reported success under ideal industrial conditions. Only when the mass of air in motion becomes substantial, as in the case of large spray booths, would it be possible to obtain good control with velocities less than 100 linear feet per minute. The smaller the quantity of air flowing and the smaller the effective range of the hood, the greater is the danger of inadequate control at velocities below 100 fpm.

If the possibility of operating small hoods at low control velocities in industrial plants is still considered promising by some students, then experiments should be conducted in the field under the influence of normal disturbances. The wide difference in industrial process environments would then be demonstrated and made a matter of record.

For the field engineer or designer, I believe it is unnecessary to fret over the viscosity or density of air and its effect on velocity contours in free space. A much greater problem will be the continual variations and fluctuations in cross drafts, thermal influence, pumping effect of passing objects, and other effects that are easily overlooked—and at the same time impossible to represent by a mathematical expression.

Ventilation Design Procedures

Surprisingly few ventilation design problems can be solved by an equation for unobstructed suction hoods, a point on which I am sure DallaValle himself will agree. One reason why this is true is that even the simplest type of exhaust hood usually has some form of obstruction squarely at its centerline, and the velocity characteristic at the centerline may be the last point that the designer needs to control. He is usually more concerned about the velocity values at the periphery of the control zone, of which the centerline intersection is a very small part.

In many process ventilation design problems the nature of the entire constant-velocity contour or surface, is of interest. Does it approximate cylindrical or spherical form, or is it so distorted or irregular that a series of flat planes is more useful in arriving at air volume requirements? Certainly the majority of designing engineers are not going to dust off their calculus to help them develop true areas of irregular velocity contours in their daily work.

At present, the geometric concept is the only reliable approach to complex enclosure and hooding problems. It does not appear that existing research data on the special conditions of unobstructed hoods is strikingly more dependable for field use than simple geometric formulas using whole-number exponents and well-known surface and plane geometrical relationships. It will be something of an educational feat if we can get most of the exhaust hood design work carried even to the point of using the common formulas for the areas of spheres and cylinders—and even these could be tabulated in any relevant handbook if the trend should make it practicable.

In my experience, the engineer, who estimates his air volumes on the basis of spherical, cylindrical, or plane surfaces of equal velocity, is sufficiently satisfied with the results to ignore the more refined procedures. Furthermore, the development of an appreciation of the shape of imaginary surfaces of equal air velocity in space is much more profitable for the average ventilating engineer than a facility with exponential equations. With such a method, he is free to build nearly any shape of hood required and arrive at the correct order of magnitude of airflow to use as the basic ventilation requirement.

Where distance between the exhaust hood face and the source of air contaminant is less than the diameter of the face (assuming a round hood), the shape of the constant-velocity surface changes so rapidly from somewhat spheroidal to a nearly plane surface that I doubt the wisdom of using the DallaValle equation in calculating conditions so close to the hood. In fact, the rapid acceleration of airflow close to the hood (within one hood face diameter) suggests that a precise equation for this condition, even if possible, would be of little practical value in industry. An operation close to

the hood opening is a sufficient disturbance to the air currents to destroy any analysis made on flow conditions without such obstruction.

Perhaps it would be of some value, in appraising the present discrepancies in laboratory results, to re-compute the constants for DallaValle's equation for each narrow increment of distance from the hood face outward. In other words, for $X/\sqrt{A}$ values of 0.5 to 1.0; 1.0 to 1.5; 1.5 to 2.0; 2.0 to 3.0; 3.0 to 5.0 and over 5.0, if such data are available.

A Simple Equation for Field Work

At the moment, I am inclined to prefer the following simple form for values of $X/\sqrt{A}$ or X/d above 1.5 or 2.0:

$$Q = CVX^2$$

I doubt the necessity of using the A in DallaValle's equation in ordinary hood design work, because the equation cannot give reliable results close to the hood face for the majority of operations that present an obstruction; and as we move away from the hood face to zones where operation obstruction will have less effect on the streamlines of flow, the influence of A rapidly becomes negligible. Probably the equation $Q = 12.6VX^2$ will faithfully represent the conditions of airflow toward suction openings when the ratio of $X/\sqrt{A}$ or X/d is above 4 or 5, and when the velocity V at the point in question is well above prevailing air disturbances. Unfortunately, the opportunity for such design conditions in practice is not frequent, and therefore we are confronted with the necessity of constructing a convenient equation that will operate in the range where X/d is 1 to 4.

I confess that, for reasons of rapid calculation, I have frequently used the following formula for demonstration as well as field purposes, without arriving at seriously erroneous airflows at distances of X greater than d for round, unflanged, suction openings:

$$Q = 10VX^2$$

And further, with the convenient minimum control velocity V of 100 fpm at normal air temperatures, the equation becomes:

$$Q = 1000X^2$$

In other words, if we want a vapor or dust controlling velocity of 100 fpm at a distance of one foot from the face of a suction tube whose diameter is not greater than one foot, we must draw 1000 cfm into the tube. This over-simplification is annoying to the scientist, but nevertheless is the kind of rapid formulation that industrial engineers like to use. They are at least within sporting range of the real target, which is also about all we can say at the moment about our theoretical analyses.

Further study of the instruments used in this project seems necessary before too many other investigators can rely on procedures that may open their data to legitimate suspicion. It is hoped that in the near future some highly sensitive, as well as durable thermo-electronic instrument suitable for field use, will make the present low-velocity devices obsolete for this kind of study.

It is increasingly evident that the best approach to the problem before us is a journey into the industrial environment to observe and measure diligently the many things that happen to process exhaust hoods. Although the fund of theoretical knowledge on airflow into exhaust openings must be augmented greatly before we can rest the case, the A.S.H.V.E. is immediately interested in data that can be applied directly in the field.

AUTHORS' CLOSURE: We are grateful for the comments on this paper. The discussions by Witheridge and Caplan need no further comment.

We agree with Mr. Poole that it is extremely difficult to measure low velocities accurately. Velocities lower than desirable were used in this study because of the

limitations of the equipment used in the study. Nevertheless, we did not measure extremely low velocities,—relatively few of them being below 200 fpm. Furthermore, we were searching for the reason for the large difference between Dalla Valle's and Silverman's findings, a difference in the order of 100 percent. Therefore, in our planning for this study we did not feel that the errors in velocity measurement and the deficiencies in this respect possessed by the instruments we selected were of concern. Of greater importance, in our opinion, was to include the instruments used by both Dalla Valle and Silverman, since our data would then point not only to the correct values but also to the reason for the difference between the earlier data. Certainly the results of our study show the wisdom of this decision.

Proper coverage of the many interesting points raised by Professor Silverman would require a lengthy discussion. Since this, in our opinion, is neither the time nor the place for a prolonged rebuttal, we will group his comments as much as practicable and reply to them very briefly.

All comments in the discussions that relate to conditions at or very close to the face of the hood are irrelevant. This was pointed out quite well by Mr. Witheridge. We tried to emphasize throughout the paper and especially in the introductory paragraph (fourth paragraph) that we were limiting our study and discussion to the area in front of the hood and at such distance from the face of it as to be out of the zone of unstable turbulence. Some of our measurements relatively close to the hood very likely may have been influenced by the unstable conditions there, but in fitting the line to the plotted results, most if not all, of this influence would be ruled out because such points would not be in line with those determined by the measurements at relatively greater distances from the hood. We confined our study to the area in front of the hood faces because in industry the sources of contamination are normally at some distance from the face of the hoods, and as pointed out by Witheridge, as we get closer to the hoods the rate of air movement increases so rapidly that contaminant capture is relatively easy.

Degree of stem immersion may have had some small influence on a few of the measurements. No attempt was made to correct for this factor for several reasons. In the large majority of the measurements, the stem was immersed almost completely. Where not *completely* immersed, the air movement over the thermometer stem varied from that at the bulb to a less but unknown value, so that correction would have been either a time-consuming procedure or a *guesstimate*. The small correction for stem immersion that would have been required in this study would have been of little significance except possibly in the *high velocity* column of Table 6. Furthermore, correction for stem immersion in this particular study is not only unimportant but also irrelevant since we were looking for a gigantic error by comparison, and were employing other instruments as well as the thermoanemometer.

In answer to Professor Silverman's questions regarding the sphere method of calibration, I might say that I do think this method is feasible and the fact that I used it answers the second part of his first question. While a very long narrow slot could be used as well as a sphere, it was not used in this study because (1) a sphere was readily available and a slot would have had to be constructed, and (2) we preferred the unidirectional flow created at all points by the sphere to the curved direction in all except one plane created by the slot.

In final conclusion, we should like to say that this study served its purpose very well by showing that (1) the relationship $Q=V(10x^2+A)$ expresses the conditions of centerline air flow into suction openings accurately enough for all practical purposes, and (2) the different results obtained by Silverman probably were due to an instrumental error,—the velocity measuring device used by him giving high values at low velocities.

1306

No. 1306

THE EFFECTS OF MOISTURE CONTENT ON THE DIFFUSION OF ODORS IN THE AIR

By RICHARD L. KUEHNER,* PH.D., YORK, PA.

PURPOSE

INVESTIGATIONS have been made on the effects of humidity on odor transmission, with the ultimate objective of controlling odors in air conditioned spaces, and it is the author's purpose to set forth the findings in this paper. The marked vagueness on the subject of odors, ranging from the very definition of an odorous substance up to and including the physiological responses accompanying smell, indicates the complexity of the entire problem.

INTRODUCTION

Because considerable difficulty was foreseen in experimentation with natural odors, both in the control of purity and in the measurement of concentrations, the studies were planned around certain synthetic or unnatural odors. To establish the validity of these investigations, it was first necessary to be assured that the test compounds lived up to the requirements of the naturally occurring odors.

The generally accepted conception on the subject is that odors are widely dispersed molecules of specific chemical substances, or, in other words, that odors, if they can be concentrated into a recognizable mass, are well defined compounds. The intangibility of odors results only from their dilute dispersion. This definition is, of course, not complete in itself, since all substances to a certain extent, have the capacity of volatilizing and suspending themselves in air, whereas relatively few possess the property of aroma. Hence, in recent years certain conditions have been added to the original definition. Grey, 1942,[1] pointed out that the animal reception of odors involves a solution process; odors

*Bacteriologist, Research Dept., York Corp.

[1]Exponent numerals refer to Bibliography.

Presented at the 53rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1947.

to be registered as such must be dissolved in the secretions bathing the olfactory areas on the nasal conchae. These secretions are lymph-like substances containing a solution of mucus, metallic salts and blood protein which render them more or less of a lipide or fat solvent. Accordingly, Dyson, 1938[2] demonstrated that all odors must be fat-soluble to a certain extent. This does not mean that they may not have water soluble characteristics, but only that they must be sufficiently fat soluble to dissolve in the olfactory secretions up to such a concentration as to elicit the physiological responses of smell. Secondly, Dyson[2] has stated that odors must possess an appreciable vapor pressure.

With these two criteria as a basis, it has been shown that odors must possess certain elemental components or molecular arrangements. Niccolini[3, 4] demonstrated that there are two major groups of elements, the olfactory positive group which consists of those elements that may form a part of at least one odorous compound, and the olfactory negative group comprising those elements which never form a part of an odorous molecule and therefore possess the power of suppressing the organileptic properties, even if combined with odorous elements.

Further work, Dyson 1929,[5] indicated that certain radicals or atomic groupings are the seat of olfactory activity in substances. Vanillal, citral, rhodinal, and benzaldehyde all contain the —CHO group, a pleasant odor. The ester group in methylsalicylate, the ether group in safrol, and the ketone group in methylheptenon are all probably the seat of their respective odors.

Stoll 1936,[6] Salomon and Meyer 1937,[7] Hornbastel 1931,[8] and Rodger and Dvolaitskaya 1937,[9] have shown that these radicals and many others are responsible for the basic characteristics of certain odors, but that type odor can be changed or modified within certain limits by the placement of prime radicals within molecules, and by the varied positions of other secondary odorous molecular radicals.

Because of the large number of possible basic molecular linkages, which can be modified by their surrounding linkages, there has been no comprehensive chemical classification of odors. Odor classification has thus far been based upon the odor characteristic itself. (It is commonly agreed that two individuals will register almost identically the quality, if not the intensity of a given odor). The most complete classification found was adopted from the Dutch Physiologist, Linnaeus, by Zwaardemaker, as outlined by Karrer.[10] Odors here have been divided into 10 different types, on the basis of their aroma. Although no attempt was made to describe modifications of basic odors, this work covers the major categories of natural and unnatural odors. As the author's work did not call for a more finite characterization, Zwaardemaker's types have been accepted. Keath,[11] Bienfang,[12] and Cramer[13] have agreed with the author's conclusions.

Since odors are normal molecules exerting appreciable vapor pressures, they must have the capacity of mixing with air by means of diffusion. The rapidity with which an odor can permeate the atmosphere in a room with a minimum of convection currents indicates that diffusion, as well as convection current, is a major factor in the dissemination of odors. In a great many instances diffusion is the apparent prime factor, since a constant source of odor imparts to the surrounding air an olfactory threshold to a distance varying with the activity of diluting convection currents. It is common knowledge that to prevent an odorous area from contaminating a non-odorous area in practical work, as in the case

of restaurant kitchens and dining rooms, certain minimum velocities of air must be maintained from the odorless area toward the odorous area, according to the intensity and nature of the polluting odor.

Accordingly, since diffusion is a major factor, can this be controlled in any way by which the spread of odorous materials in air streams could be controlled? Woedemann 1935[14] claimed that odor is much stronger in dry air than where the air is relatively more humid; Rosenau in 1940[15] claimed the reverse by stating that odors were much stronger in moist air than in dry. The author's prime objective then was to attempt to test these two conflicting statements on the possibility that, by controlling humidity, odor spread could thereby be controlled.

EXPERIMENTAL APPARATUS

Because odor diffusion appeared more easily measurable than convective distribution of odor, the author's experiments were established around this phenom-

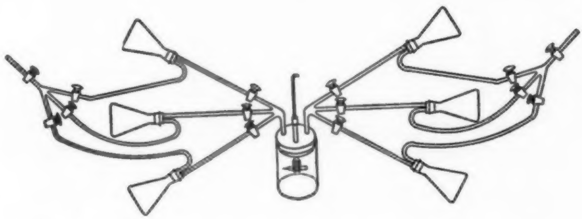


FIG. 1. FIRST DESIGN OF ODOR DIFFUSION APPARATUS

enon. Quite simply, it was planned to inject measured quantities of various odors into a vapor tight air space of controlled temperature and humidity. The presence or absence of the odor could then be determined at certain fixed distances from the source of injection at various time intervals. The differences in times required for the various odors to cover these distances under varied conditions of temperature and humidity would then be recorded and compared, the human sense of smell being used as the odor measurement.

In the first attempts, diffusion apparatus No. 1 was constructed as shown in Fig. 1. This was simply a central 125 ml beaker serving as the odor reservoir. On the stoppered end of this were placed two glass manifolds, each being connected by rubber tubing of ¼ in. I.D. to three 125 ml at Erlenmeyer flasks equal distance from the central reservoir. The odor was released in the reservoir and at varying time intervals each flask was tested by sense of smell, in succession, for the presence of the odor. However, reproducible results could not be obtained and in many instances the odors failed to migrate in periods as long as 15 hr.

Diffusion apparatus No. 2 was then designed and constructed as indicated in Fig. 2. This was a Pyrex glass tube of 4 ft 7½ in. overall length (conveniently flanged for cleaning) having an internal diameter of 3 in. At one end were

located three tubes which could be closed by stopcocks. Original air and make-up air were admitted through a tube, across the internal discharge end of which was placed a glass flange for preventing the jet injection of air, thereby minimizing the possibility of convection currents. The apparatus was evacuated through stopcock B; a manometer was attached to tube C for giving continuous readings on the pressure in the system. At the opposite end of the apparatus was placed a single tube D for the injection of odors.

Distributed at distances of 3, 6, 18, and 30 in. from the odor injection point were placed four sampling connections (E, F, G, and H). These comprised simple glass tubes of ⅜ in. I.D. projecting into the true center of the diffusion vessel. At this point the tubes were bent at right angles and extended for a distance of ½ in. in order that the internal sampling orifice might be in the same

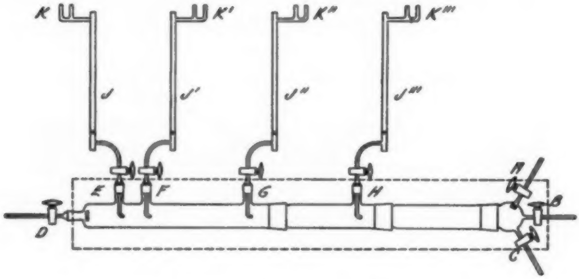


FIG. 2. DIFFUSION APPARATUS USED IN TESTS

plane as the movement of the odor. These connections were equipped with stopcocks and rotameters J, J′ and J″ on the ends of which were fixed nosepieces K, K′ and K″.

This entire apparatus, excluding the open end of the odor injection tube and the four sampling connections, was immersed into a water bath for controlling temperatures (dotted lines in Fig. 2).

To standardize the conditions, it was found necessary to develop a technique for purifying the experimental air. The apparatus involved is shown in Fig. 3. Air at 300 lb pressure was brought in through a Mandler bacteriological filter L, a packed wool filter M, and an activated carbon cylinder N, the latter two being surrounded with solidified carbon dioxide. The air was then passed through tube O to the diffusion apparatus No. 2. Humidity was determined by discharging part of the air over dry bulb thermometer P and wet bulb thermometer Q to the atmosphere R. A constant pressure could be held in tube O by the regulation of the valve on the discharge orifice S.

For the high relative humidity experiments, this purified air was humidified by the apparatus shown in Fig. 4. The air from tube O on Fig. 3 was supersaturated with steam produced in flask T. The air was then cooled to bath

temperature by condenser U, the entrained moisture being removed by the three traps V. The traps were immersed in the water bath containing the diffusion apparatus No. 2, and the condenser was supplied with the same water by pump W.

EXPERIMENTAL PROCEDURE

The experimental odors were chosen in accordance with the foregoing criteria. In the preliminary screening, the compounds must conform to Dyson's and

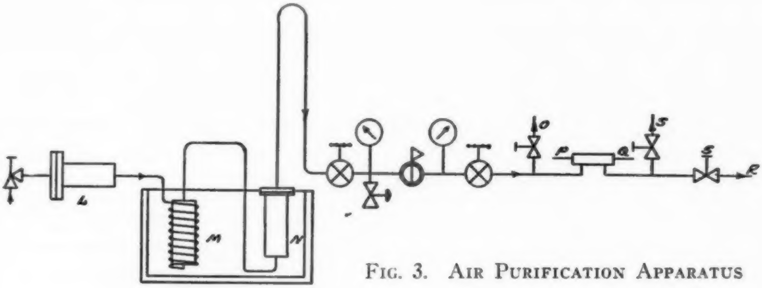


FIG. 3. AIR PURIFICATION APPARATUS

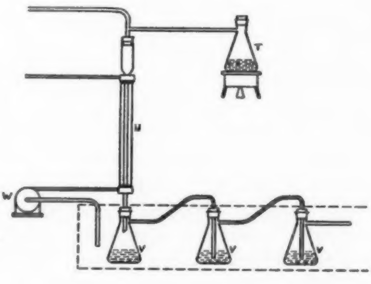


FIG. 4. APPARATUS FOR HUMIDIFYING PURIFIED AIR

Niccolini's stipulations. At this point, position in Zwaardemaker's classification was taken into account in an attempt to cover odor classes rather than individuals. The final selection was made on the basis of occurrence in the air conditioning industry. With this procedure, five odors given in Table 1 were then decided upon.

It must be pointed out that each of the general odor types listed in Table 1, especially body and tobacco smoke odors, is an aromatic aggregate (the author has found 7 in the former and 28 in the latter to date). Therefore, it is not claimed that each of the specific compounds listed is the one responsible in each aggregate, but rather, that each is a major component responsible for only a part of the overall effect of the aggregate.

It was understood that, before an experimental procedure could be worked out, there were certain critical points in apparatus and technique which might very well add variables to the interpretation of results.

From the beginning, the reliability of the human nose for a measurement means was a critical point in this work. However, it was found that the sensitivity of the nose did not decrease appreciably upon repeated smelling if the odor was smelled only at threshold or slightly above. If a very high concentration of odor was smelled, the olfactory mechanism was temporarily paralyzed in relation to that specific odor and did not return to 100 percent efficiency for some considerable time thereafter. If the smelling was only at threshold, E. C. Crocker's findings[15] were confirmed that the nose would return to approximately 100 percent efficiency after the passage of about 5 sec time. This was not true, however, with irritating type odors; it was found that even threshold samples of irritant odors depressed the efficiency of the nose in relation to any odor for a period of several hours. Since this experimentation was designed for work only at threshold, and no irritants were included, the nose was considered to be sufficiently reliable for this purpose if all samples were identical in quantity, pressure and velocity. Subsequent experimentation has borne out this assumption.

TABLE 1. CLASSIFICATION OF ODORS

ODOR	ZWAARDEMAKER'S CLASSIFICATION	NATURAL OCCURRENCE
1. Acetic acid	Etheral Type I	human body odor
2. Methyl acetate	Aromatic Type II	decomposition of fruits and flowers
3. Iso-valeric acid	Rancid Type VIII	animal sweat and burning tobacco
4. Butyric acid	Rancid Type VIII	dairy products
5. Pyridine	Narcotic Type IX	protein decomposition and tobacco odor

In getting better control of quantity and velocity of samples, a constant internal pressure superseded respiratory suction as the motive force behind the sample. To obtain a satisfactory sample, the odor-air mixture from the diffusion chamber must be forced through the rotameter, at the sample connection, at the rate of 4 liters per minute for a period of 3 sec. This gave the required 25 ml to replace the residual air from the previous samples in the rotameter system, and an additional 175 ml of sample for smelling. To obtain this velocity-quantity, it was necessary to hold the overall pressure within the apparatus at not less than 4 in. of water. Respiratory exhaustion of a sample from the diffusive vessel at atmospheric pressure could not be satisfactorily controlled for duplicating experiments.

To be assured that results of diffusion at elevated pressures were comparable with those at normal atmospheric pressures, duplicating odor diffusion experiments were run on acetic acid at 4, 6, and 10 in. of water pressure, and a final one at atmospheric pressure using respiratory exhaustion for the sample. In all cases the results were identical.

Another critical point was that of the injection of odor. The technique was simply that of injecting into the system by capillary pipette that quantity of liquid odor which would be vaporized instantly. It was found that 0.009 ml

of isovaleric acid, the least volatile of the five odors, would instantly flash to gas when injected at 60 F. This quantity was used for each of the odors in this experimentation. However, in some instances precise control of this quantity was impossible. To determine the effect of variable injection quantities, duplicating experiments were run with isovaleric acid and acetic acid. In the first series 0.009 ml of odor was injected and in the second 0.0045 ml was injected. In both cases it was found that a marked change in the quantity of odor did not appreciably affect the speeds of diffusion. It was therefore concluded that slight differences in quantity of injection would have no effect upon the rates of diffusion.

After thorough washing, drying and assembling, the apparatus was sterilized and deodorized by blowing out with steam at 5 lb pressure for 10 min. The system was then evacuated to 100 μ (microns) of mercury, filled with dry air, and re-evacuated to 100 μ. It was then refilled with the type of air with which the experiment was to be run, whether humidified or dry air. With the purifying apparatus described it was possible to clear the air of all odors and of all suspended materials of 1/10 μ size and below. The air was likewise dried to 0.2002×10^{-5} grains of moisture per pound of dry air. This will later be referred to as 0 percent relative humidity. In the humidification system in this experiment, air was considered to approximate saturation or a relative humidity of almost 100 percent. After a 15 min washing period with this air, the relative humidity was checked with the dew point apparatus. The contents of the system were allowed to rest for a period of 15 min at atmospheric pressure, while the temperature was being adjusted. The odor was then injected and the pressure immediately adjusted to 4 in. of water.

A pilot run was made in which the operator smelled first sample connection E every 15 sec until the appearance of the odor. He then moved to sample connection F to determine the presence or absence of odor at this point. The operator then advanced to connections G and H, Fig. 2, regulating the sampling intervals according to the time involved for the passage of odor between the two immediately preceding sample connections. It was a rule that the odor must be recognizable as a specific entity before it was recorded as positive. Two duplicating runs were considered as a reliable result and were recorded as such.

After the completion of a run, the apparatus was re-steamed, and the preparations were repeated as described previously.

EXPERIMENTAL RESULTS

These results on the diffusiveness of odors have been tabulated at 0 percent and 100 percent RH in Tables 2 and 3, respectively. They were prepared by selecting and averaging two or three results of reproducible runs on each of the odors under the conditions studied. For runs to be termed as reproducible, the time of the detection of the odor at each of the four sample connections must not vary from the time of former runs by more than $\pm$ 10 sec. This has been considered well within the experimental error of the human nose.

Readings taken at point G, Fig. 2, were most reliable, and therefore were used in plotting Figs. 5 and 6, which show results for the two air temperatures 60 F and 110 F, respectively, and for 0 and 100 percent RH in both cases.

By means of the one dimensional diffusion equation, the validity of the experiments has been checked as to their being a true diffusive phenomenon. By solving this equation it has been found that a plot of the log of the time of travel against the square of the distance over time of travel should be linear and should produce a curve sloping downward toward the right. In plotting these data with such axes, it has been found that the curves shown sloped upward toward the right. However, to swing the lower end of any of the lines into theoretically proper position would require such a small increment of time (15 sec in most

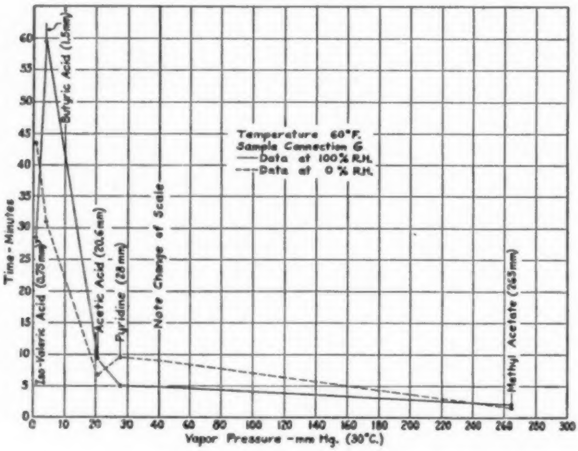


FIG. 5. RELATION OF RATE OF ODOR DIFFUSION, AT 60 F AIR TEMPERATURE TO VAPOR PRESSURE OF SUBSTANCE

cases) that the results have been considered to be specifically diffusive. Although this shows that the diffusion of the odors given under actual conditions is slower than theory would indicate, it can be readily accounted for by the added friction involved.

It will be noted, in both Figs. 5 and 6 and in Tables 2 and 3, that the diffusion rates of odors vary in the main according to the vapor pressure gradient. Hence, the rate of speed of a given odor varies with its absolute vapor pressure. This is indicated by the general downward swing of the lines, on both graphs in Figs. 5 and 6, from left to right. Similarly, the higher the temperature and the higher the concentration of the odorous vapor, the greater the speed of diffusion.

It is appreciated that these findings are at variance with the results of the preliminary experiments on the injection of odor, in which it was found that reduction of the initial odor charge by one half showed no effect on the rate of

diffusion. However, it would appear that in the former case the author's experiments were well above that critical point at which a change in concentration would result in a comparable change in the rate of diffusion.

The results in this paper indicate also that relative humidity plays some minor role in the diffusivity of the test compounds. Tables 2 and 3 show a general difference in the various comparable points between 0 percent and 100 percent RH, these differences being more marked at 60 F than at 100 F. It will be noted that, in the case of pyridine and also isovaleric acid, the presence of a high

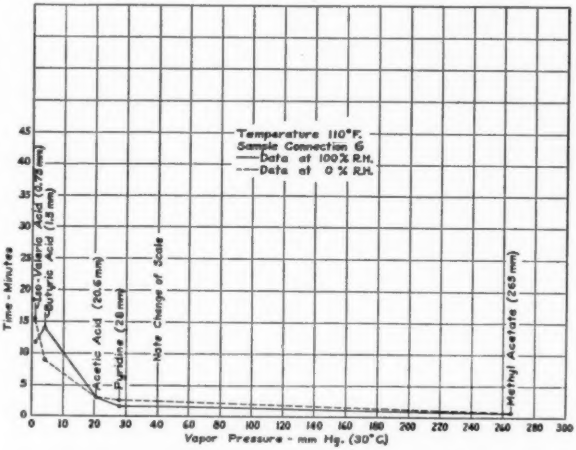


FIG. 6. RELATION OF RATE OF ODOR DIFFUSION, AT 110 F AIR TEMPERATURE TO VAPOR PRESSURE OF SUBSTANCE

relative humidity in the testing air menstrum seems to increase the rate of diffusivity of these two compounds. Here again the difference is greater at the low temperature of 60 F. However, in the case of the other three—acetic acid, butyric acid and methyl acetate—the presence of a high relative humidity seems to depress the velocity of diffusion, the odor traveling faster under conditions of 0 percent RH. In the case of acetic acid, the odor travels from the first two sample connections at a greater rate under 0 percent RH, and then slows up at the third orifice.

It was originally thought that any differences in speed resulting from change in only one variable (relative humidity) might be accounted for on the basis of the solubility of the various odors. It will be noted, however, that in the one group acetic acid and butyric acid are soluble in all proportions in water, while methyl acetate is soluble only to the extent of 31.9 g (grams) per 100 ml. of water at 20 C. In the other group, pyridine is soluble in all proportions of water, while isovaleric acid is only soluble to the extent of 4.2 g per 100 ml of

TABLE 2. TIMES OF ODOR DIFFUSION AT 0 PERCENT RH

Odor	Percent R.H.	Sample Connection	60 F		110 F	
			Min	Sec	Min	Sec
Pyridine	0	E (3 in.)	0	53	0	20
		F (6 in.)	2	45	0	37
		G (18 in.)	9	30	2	30
		H (30 in.)	23	30	8	0
Isovaleric Acid	0	E	1	45	1	0
		F	4	45	3	45
		G	43	0	15	30
		H	....	....	31	0
Acetic Acid	0	E	0	40	0	23
		F	1	36	0	57
		G	6	45	3	0
		H	....	....	12	0
Butyric Acid	0	E	2	24	0	15
		F	8	0	1	54
		G	31	0	9	0
		H	....	....	....	....
Methyl Acetate	0	E	0	10	0	7
		F	0	21	0	13
		G	1	18	0	44
		H	....	....	....	....

TABLE 3. TIMES OF ODOR DIFFUSION AT 100 PERCENT RH

Odor	Percent R.H.	Sample Connection	60 F		110 F	
			Min	Sec	Min	Sec
Pyridine	100	E (3 in.)	0	14	0	13
		F (6 in.)	0	45	0	22
		G (18 in.)	5	0	1	45
		H (30 in.)	16	15	....	....
Isovaleric Acid	100	E	0	52	0	45
		F	3	15	2	15
		G	28	30	11	45
		H	....	....	....	....
Acetic Acid	100	E	0	30	0	10
		F	1	36	32	0
		G	9	15	3	0
		H	18	30	....	....
Butyric Acid	100	E	6	15	1	24
		F	19	0	4	15
		G	59	30	14	15
		H	90	0	....	....
Methyle Acetate	100	E	0	10	0	11
		F	0	20	0	23
		G	1	45	0	42
		H	4	45	....	....

water at 20 C. Therefore, solubility alone does not appear to be directly responsible for differences in the effect of relative humidity upon the diffusion of odors.

In general, then the basic vapor pressure gradient of the various substances under experiment is indicated as the major fundamental force governing the speed of odor diffusion. As anticipated, the speeds are markedly changed by any alteration in temperature within a given sample, but in relationships between the various other substances, the general slope is still retained as determined by the vapor pressure gradient. This is true whether 100 percent or 0 percent RH is employed. It will also be noted that during the experimentation on low vapor pressure compounds the presence of 100 percent RH seems to have a slowing effect upon the rate of diffusion, whereas, among those of higher vapor pressures the lines reverse themselves in the presence of a higher humidity, and the presence of the higher relative humidity seems to increase slightly the rate of diffusion. However, the reversal in the case of isovaleric acid contradicts this conclusion. At any rate, these differences seem to be so slight, especially at the higher temperature of 110 F, that many of the various points could not be defined as being separate and distinct within the limits of the accuracy of the author's technique.

CONCLUSIONS

It has been shown that the spread of odors, and thereby the intensity of odors in air conditioning systems, cannot be controlled with any degree of success by the simple control of relative humidity.

Although these data indicate that the relative humidity of the atmosphere does exert an effect on the ability of odors to permeate that atmosphere, this is not a predictable effect since three substances were diffused more rapidly at 100 percent RH than at 0 percent while in the case of the other two, the reverse was true. Accordingly, since most odor problems involve odor aggregates, one could not expect a uniformly depressant effect with either high or low relative humidity. Secondly, the differences in rates of diffusion of specific odors at extremes of from almost 0 percent to almost 100 percent RH are so small that commercial design relative humidities of from 45 to 55 percent would have no effect on the odor problem.

Similarly, commercial design temperatures which vary from 75 deg to 85 deg dry bulb can have practically no effect on the odor problem, in varying only 10 deg.

It has been concluded, further, that the rate of odor diffusion is controlled primarily by the vapor pressure gradient of the substance; that, within a given odor, the absolute speed of transmission of that odor, as controlled by the vapor pressure gradient may be varied markedly by the temperature at which the phenomenon takes place. However, this is not a hard and fast rule, since marked variants from this rule have been found. These variations cannot be satisfactorily explained by any of the factors which were considered to play a part in diffusions of this sort. No correlation was found between velocities of diffusion and the known compound characteristics, such as water solubility, olfactory thresholds, and molecular areas. Accordingly, predictions cannot be

made concerning the diffusibility of unknown odors without taking into account certain, as yet unknown, factors.

BIBLIOGRAPHY

1. Anatomy of the Human Body, by H. Gray. (Lea & Sebiger Publishing Co., N. Y., 1942.)

2. The Scientific Basis of Odor, by G. M. Dyson (*Chemistry & Industry*, 1938, 647-51.)

3. A Scheme for the Subdivision of the Chemical Elements with Regard to their Olfactosensibility, by P. M. Niccolini. (Boll. Soc. Ital. Biol. sper Vol. 9, 1934, 369-70.)

4. Physico-pharmacological Study of Odor, by P. Niccolini. (Arch. Ital. Sci. Farmacol Vol. 6, 1937, 241-78.)

5. Some Aspects of the Vibration Theory of Odor, by G. Malcolm Dyson. (Perfumery Essential Oil Record, Vol. 19, 1928, 456-9.)

6. Many-membered Rings & Must Odor, by M. Stoll. (Drug & Cosmetic Industry Vol. 38, 1936, 336.)

7. Survey of the Properties of Many-membered Cyclic Imines, by L. Ruzicka, G. Salomon and K. E. Meyer. (Helv. Chim. Acta, Vol. 20, 1937, 109-28.)

8. Odor Intensity, by E. M. V. Hornbastel. (Arch. Ges. Physiols—Pflugers—Vol. 227, 1931, 517-38.)

9. Odor of Hyptyl Esters, by M. Roger and F. Dvolaitskaya. (Recherches, Vol. 1, .937, 13-15.)

10. Organic Chemistry, by Paul Karrer. (Nordemann, N. Y., 1938.)

11. Standardization of Odor and Flavor, by B. Keath. (Perfumery Essential Oil Record, Vol. 28, 1937, 52-4.)

12. Dimensional Characterization of Odors, by Ralph Bienfang. (Chron. Botan. Vol. 6, 1941, 249.)

13. Odors, Olfaction & Trained Odor Detectors, by M. Cramer. (Protar, Vol. 7, 1941, 127-33.)

14. Influence of the Water Vapor Content of an Odorous Gas on Olfactory Sensation, by H. Woedemann. (Arch. Neerland Physiol. Vol. 20, 1935, 541-5.)

15. Preventive Medicine and Hygiene, by M. J. Rosenau. (D. Appleton-Century Co., N. Y., 1940.)

16. Flavor, by E. C. Crocker. (McGraw Hill Co., N. Y., 1945.)

DISCUSSION

W. L. FLEISHER, New York, N. Y. (WRITTEN): The conclusions drawn from this paper, presented by Dr. Kuehner, were indeed gratifying, so far as my own reactions are concerned. The world is so filled with people who accept hearsay as fact that the amount of rubbish spoken and published about humidity in any form has been one of the dominant misconceptions in all air problems.

It has been my contention, as many people here know, that the proper moisture content of the air is one of the fundamental and basic requirements for health and comfort. There has been so much unfavorable comment on relative humidity as an adjunct to air conditioning, that it is interesting to find, at last, a group of investigators who categorically state that the relative humidity of the air is not a function of odor

diffusion; that, of the five basic odors, three seem to diffuse better in 100 percent saturated air, and two better in 0 saturation, and that actually the diffusion of odors is, to a great extent, or so far as the average person can discern, independent of the relative humidity.

There are so many groups who either condemn the use of moisture in the air to increase comfort due to various factors affecting material objects, rather than the psychological and electrical equilibrium of the people, that it is gratifying to find fallacious at least one point which has always been cited against proper humidification.

C. S. LEOPOLD, Philadelphia, Pa.: I know we are all very appreciative of Dr. Kuehner's excellent paper on rate of odor diffusion, but I fail to see why the rate of diffusion is of primary importance in a ventilated structure. The minimum air movement which can be expected in an ordinary system is from 10 to 5 fpm. If you attempt to produce a lesser air motion, you will encounter thermally induced air currents which are in excess of this figure. The rate of diffusion in still air does not appear to be a good criterion for odor or water vapor.

You could, I believe, place silica gel in the central flask of the apparatus shown in Fig. 1 and find that with open cocks, a long, long time would elapse before you would collect any appreciable moisture. Within our understanding, the convection noted is more rapid than diffusion.

You mentioned two references. I should like to refer to a third, prepared by Prof. R. M. Smock,† Cornell University, on elimination of odor in apple storage. In this work, he chemically measured the odor-producing substance of wood. The emanation was found to be, roughly, in direct proportion to the relative humidity.

At the point of being indelicate, I should like to present a practical aspect of this problem and assure you that a group of people are a lot more appetizing within a cool, dry atmosphere than they are in a hot, moist one. This effect may be ascribed to effective temperature, if you will, but certainly a group of people produce more odor under warm, than under cool conditions. I am not referring to the minor variations shown in this paper.

One other question, which I suspect is covered in the literature, although I have never found it, is, whether the sensitivity of the nose itself is affected by temperature and relative humidity. If it is, then in this experiment we cannot compare the extremes, because our instrument of measurement has varied.

L. E. SEELEY, Durham, N. H.: The account of this experiment reminded me of some work I did at one time, and I would just like to say that obtaining either 0 percent RH or 100 percent is an exceedingly difficult job in itself, and I was further reminded that it appears to be impossible to take either a 100 percent or 0 percent sample and conduct it through a rubber tubing without having the condition changed. The hydroscopic properties of the rubber will not permit the sample to remain unchanged. I was not sure from the account just how much rubber might have been employed somewhere in the system.

AUTHOR'S CLOSURE: Replying to Dean Seeley first, it was found out that it is difficult to get 0 and 100 percent relative humidity. As indicated in the presentation, we do not pretend that we got that actually, but just for the simplicity of speaking, we refer to it as such.

Secondly, in the rubber tubing—that is one of the explanations for our failure in the first diffusion apparatus. We found you could not repeat experiments in an apparatus which contained rubber tubing because of absorption. You could not remove the traces of odor with rubber tubing even with vigorous processes, like dichromate cleaning solution. That was quite expensive and time-consuming.

Now, as far as the rate of odor diffusion not being applicable to the problem, I want to point out two practical evidences which we feel support our theory that this is applicable. Take for example, a room wherein you have an odor source. Let us say a person. He may not be sufficient to impart odor intensity to the entire room, but you will avoid sitting beside him and you will avoid associating with him.

†Studies on Odor Elimination in Apple Storage, by C. R. Gross and R. M. Smock. (*Refrigerating Engineering*, December 1945.)

Secondly, in practical work we have found in such places as restaurants and kitchens and dining rooms, you must have a certain minimum velocity of air passing from the dining room to the kitchen to keep the odors from migrating back against this air velocity and getting into your dining room and affecting the eaters.

I seem to recall that you must have a minimum velocity of 200 fpm to overcome the diffusive action of cooking odors.

As far as relative humidity and temperature affecting the human nose, we have been able to find no evidence that there is such an effect. We found, though, on high relative humidity days and low relative humidity days, as well as on high temperature and low temperature days we got no variation in our results from the chronic effect of the normal atmosphere on the nose.

1307

No. 1307

DEHUMIDIFICATION METHODS AND APPLICATIONS

By John Everetts, Jr.,* San Francisco, Calif.

PRIOR to World War II, the use of sorbents in the air conditioning industry was purely secondary, with the exception of a few isolated and specialized applications.

During the war, the use of sorbents for the dehumidification of air received a tremendous impetus for these reasons: (1) the preservation of materiel and (2) the increased number of manufacturing plants requiring humidity control to improve the quality and quantity of their product.

Manufacturers, as well as shippers, soon recognized the heavy toll that moisture in the atmosphere was taking through corrosion and tarnishing of metals; changes in dielectric properties of radio and electronic equipment; and mildew, mold and fungus rot in foods, clothing, leather, and many other materials. A large portion of this damage occurred before the materials were shipped out of the United States. To prevent this damage, it was necessary to package the materials in vapor-proof wrappings, and to include a small container of sorbent material to assure the preservation of the product. Where the product is stored in bulk, it is done in dehumidified warehouses or storerooms where the humidity is definitely controlled. The Navy, in its post-war program, is now completing the dehumidification of approximately 3000 vessels in the inactive reserve fleet. The Navy is also dehumidifying a number of their large warehouses to preserve the dielectric properties of radio, radar, sonar, and other electronic equipment and spare parts. The Army has taken similar measures to preserve tanks, ordnance equipment, and other valuable material. Today, industry, both here and abroad, is recognizing the importance and economic value of dehumidification for preservation, as well as its use in manufacturing processes.

There are two general methods for dehumidifying, namely, *static* and *dynamic*. Static dehumidification is accomplished by merely placing a container of sorbent

*Consulting Engineer. Member of A.S.H.V.E.
Presented at the 53rd Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January 1947.

material inside the package, or the enclosure which is to be treated. The package or enclosure should be reasonably tight to prevent the infiltration of moisture-laden air. In some applications it is also necessary to vapor-proof the enclosure

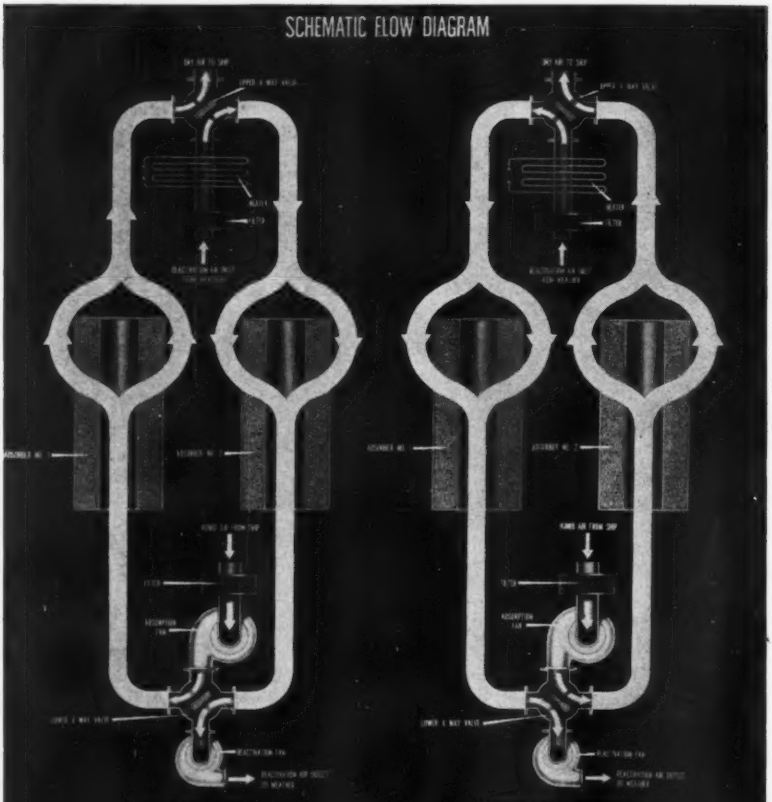


FIG. 1. SCHEMATIC FLOW DIAGRAM FOR CONTINUOUS OPERATION OF DEHUMIDIFICATION UNIT

to prevent vapor infiltration. The sorbent may be discarded after use, or it may be reactivated and used over again according to the type of sorbent used and the application. Most solid sorbents can be reactivated by placing them in an ordinary oven and heating to a temperature of 250-300 F for a few hours. Obviously, it is impossible to control the relative humidity at any one point with static

dehumidification. The use of an indicator gel will give a visual approximation of the conditions. If the relative humidity is low, the indicator gel will show a blue color and, as the relative humidity increases, the indicator gel will turn pink. As soon as the indicator starts to show a pink color, it is necessary to replace the sorbent or to reactivate the one in use. The quantity of sorbent material required will vary widely, according to the product to be preserved and

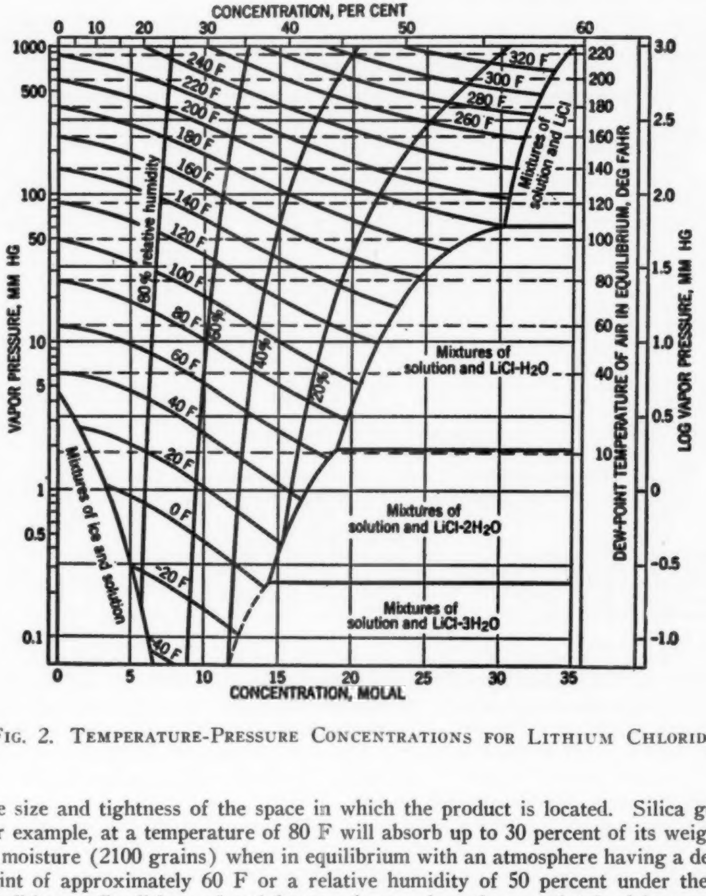


FIG. 2. TEMPERATURE-PRESSURE CONCENTRATIONS FOR LITHIUM CHLORIDE

the size and tightness of the space in which the product is located. Silica gel, for example, at a temperature of 80 F will absorb up to 30 percent of its weight in moisture (2100 grains) when in equilibrium with an atmosphere having a dew point of approximately 60 F or a relative humidity of 50 percent under these conditions. Conditions of satisfactory dryness have been maintained in empty fuel tanks over long periods of time with as little as one pound of silica gel per 100 cu ft of volume, after the initial drying has been completed. There are many applications where static dehumidification is a necessary requirement for

the proper preservation of material, either in storage or in transit. Each application, of course, requires individual study to determine the most economical and serviceable arrangement. Solid sorbents which do not change their physical characteristics are the only type which can be used for static dehumidification.

Dynamic dehumidification is obtained by circulating air through a sorbent material which is automatically reactivated to maintain continuous operation and hold a predetermined constant humidity condition within the treated space. A schematic flow diagram of a continuous-operation dehumidification unit is shown

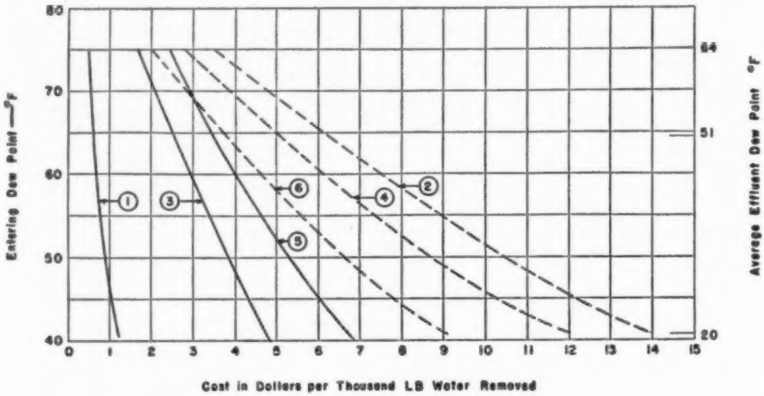


FIG. 3. OPERATING COST COMPARISON OF SORBENTS *vs.* REFRIGERATION IF SORBENT IS REACTIVATED BY STEAM

Conditions
Entering dry bulb—95 F
Refrigeration electrically operated
Sorbent reactivated by steam
Electric power 1c. per kwhr
Steam $0.30 per 1000 lb

1. Sorbent with 135 F effluent air
2. Refrigeration with 135 F effluent air
3. Sorbent with 95 F effluent air
4. Refrigeration with 95 F effluent air
5. Sorbent with 70 F effluent air
6. Refrigeration with 70 F effluent air

in Fig. 1. Dynamic dehumidification is used where a definite humidity condition must be maintained, and in applications which are too large for the economical use of the static method. The research work which was done during the war for the Navy at the Engineering Experimental Station, Pennsylvania State College, has resulted in the development of machines which are more efficient and serviceable than the pre-war models.

During the war, sorbent materials were improved in quality as the result of greatly increased production and more rigid specifications. In the solid adsorbents, silica gel, activated alumina, and activated Bauxites were used for static dehumidification; silica gel and activated alumina were used in the dynamic dehumidification machines. Among the liquid sorbents, lithium chloride, of the halogen salts group, and glycols were important in dynamic dehumidification.

The advantages of solid or liquid sorbents are dependent upon the requirements of their application, and the economic analysis of the available utilities involved in the operating expense. The dew-point depression of a solid sorbent is the function of the *temperature* of the sorbent material, whereas for the liquid sorbents, the dew point depression is limited by the *density* of the solution (see Fig. 2).

Liquid absorbents are usually in the form of hydrates, and such being the case, the density must be kept above the freezing or crystallization point of the solution. As the temperature is lowered, the operating range is reduced, and, at the same time, the viscosity of the solution is rapidly increased, thereby adding to the power consumption in the pumping cycle.

The sorption characteristics of hydrated solutions also vary where there is a change in phase of the sorbent. For example, solid calcium chloride has a greater sorption ability than has the liquid form.

Solid adsorbents have no phase-change characteristics, and therefore can be used over a much wider range of conditions.

A system using solid sorbents has certain advantages over one using liquid sorbents:

1. Simplicity of design and operation.
2. No periodic replacement of sorbent material because of loss.
3. Psychrometric conditions are not affected by the inclusion of air borne dust, dirt, and other impurities.
4. No corrosion hazard.
5. Applicable for portable and mobile units.
6. No carryover of sorbent in air stream that may be deleterious to product being processed.

The advantages of a system using liquid sorbents over that of a system using solid sorbents may be listed as follows:

1. Uniform conditions of temperature and humidity of delivered air.
2. Humidification when required.
3. Removal of mold spore and bacteria from the air passing through the sorbent.

These advantages apply to those sorbents which are at present commercially available for the prime purpose of dehumidifying air. Certain clay and earth-type solid sorbents will break down into fine dust through excessive handling or vibration and will cause the bed to pack to a greater density, thereby increasing the resistance to the air flow. Particles of this dust carried into the air stream are objectionable. Corrosion from liquid sorbents may be greatly reduced by the proper treatment of the solution and the selection of the least corrosive types. For example, zinc chloride is one of the best sorbents in the halogen group; however, it is so highly corrosive, particularly in the regeneration cycle, that it is commercially impractical. Carbitol, a glycol derivative, will theoretically give a 90 deg dew-point depression from atmospheric conditions, but because of the complicated and expensive method required for regeneration, it, too, becomes impractical at the present time. Calcium chloride has little value in this field. It has low sorption qualities, is highly corrosive, and is difficult to handle. Its

low cost is offset largely by the additional quantity required to meet most of the conditions within its limited application.

There has been considerable discussion in recent years as to the relative value of mechanical refrigeration for dehumidification applications. One of the proposed methods is to use a cooling coil to reduce the air conditions to the desired dew-point, and then a reheat coil with the hot refrigerant gas to raise the air temperature to the desired dry bulb. In effect, this would be an air-cooled condenser. If electrical energy is used to regenerate the sorbent, as well as operate

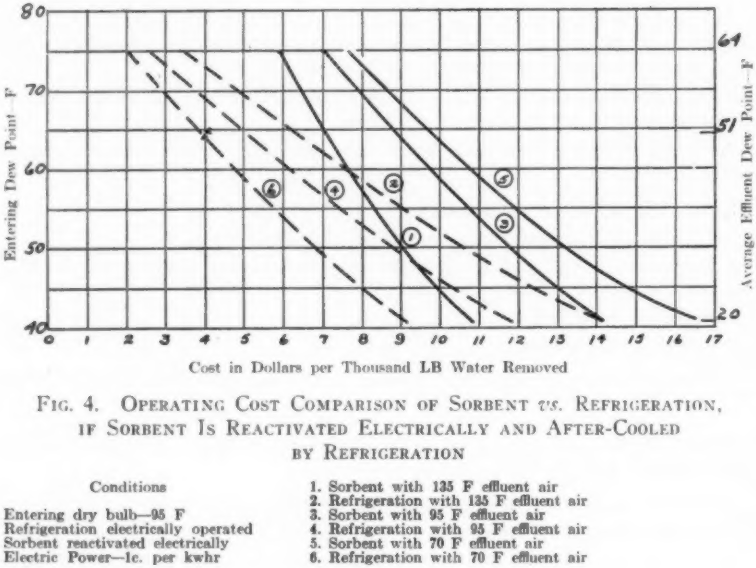


FIG. 4. OPERATING COST COMPARISON OF SORBENT *vs.* REFRIGERATION, IF SORBENT IS REACTIVATED ELECTRICALLY AND AFTER-COOLED BY REFRIGERATION

Conditions	
	1. Sorbent with 135 F effluent air
Entering dry bulb—95 F	2. Refrigeration with 135 F effluent air
Refrigeration electrically operated	3. Sorbent with 95 F effluent air
Sorbent reactivated electrically	4. Refrigeration with 95 F effluent air
Electric Power—1c. per kwhr	5. Sorbent with 70 F effluent air
	6. Refrigeration with 70 F effluent air

the mechanical refrigeration equipment, then refrigeration is favored over a rather wide range of dehumidification applications. However, if steam is used for regeneration of the sorbent, and electric power is most practical for the refrigeration equipment, then dehumidification by sorbents is the most economical method. Fig. 3 shows a comparison, in operation costs, of dehumidifying by sorbents, using steam for regeneration and with supplementary electrically operated mechanical refrigeration where required, versus the use of mechanical refrigeration only. It will be noted that, under the conditions given, mechanical refrigeration is more economical than the sorbent method, only when the delivered dry bulb is 70 or lower and the entering dew points are above 65 F. Under all other conditions, the sorbent method is shown to be more economical. The greater spread between the delivered dry bulb and delivered dew-point favors the sorbent method and affects mechanical refrigeration adversely, be-

cause in the latter case it is necessary to reduce the suction pressure, thus reducing compressor capacity and at the same time increasing the condensing temperature. This increases the power input to the compressor. As the difference between the dry bulb and the dew point is reduced, this cumulative effect is reduced in favor of refrigeration and against sorbents, because with sorbents it is necessary to increase the after-cooling requirements to meet the conditions. Fig. 4 shows a direct comparison where electric regeneration is used. As most of the industrial and commercial applications fall within the limits shown in Figs. 3 and 4, the use of mechanical refrigeration is not very promising for dehumidification unless steam is not available for sorbent regeneration. Other charts may be made showing comparisons for power, steam, or gas rates other than the values shown in these charts. The operating cost for sorbents may be further reduced from that shown, by using after-cooling at the higher temperatures through the medium of cooling tower water, well water, or partial evaporative cooling.

The data from which the curves in Figs. 3 and 4 were determined were taken from actual tests of dehumidification machines using solid sorbents. Mechanical refrigeration data were taken from performance tables of representative manufacturers of this type of equipment. The values shown will vary slightly with the variations in the many types of cooling coils, condensing units, and the type of solid sorbent used, so that, these curves should be considered as a qualitative, rather than a quantitative analysis. Each application will require its own analysis, predicated upon utility rates current with the job, and upon the characteristics of each type of machine. The values for liquid sorbents will fall within the range shown for the solid sorbents, if comparable systems are used. The total load for all sorbents is practically the same, except for the minor difference in the heat of absorption, which will vary slightly according to the type and kind of sorbent material used.

A third method has been proposed, and in a few instances used, to prevent condensation of moisture on materials stored in warehouses. This method employs the use of heat to raise the dry bulb temperature in the space, and reduces the relative humidity to the desired level. This is not dehumidification in any sense of the word, because moisture has not been removed from the air. There are several objections to this method:

1. On a hot, humid day, the temperature required in the space is higher than that allowed for employees working in such spaces.
2. This method does not prevent condensation of moisture on windows, beams, or other appurtenances of high conductivity, if the outside temperature should suddenly drop below the inside dew point.
3. In many cases, the higher temperatures required will have a deleterious effect upon the product stored.
4. The operating cost is much greater than the dehumidification to the same relative humidity condition.

Table 1 shows the requirements for a warehouse of standard construction size 100 ft × 40 ft × 15 ft. The steam required for heating is based upon the heat loss through the building, plus one air change per hour.

The steam required for dehumidification is based upon the removal of moisture from the outside dew-point to that dew-point corresponding to the desired rela-

TABLE 1. REQUIREMENTS FOR A WAREHOUSE OF STANDARD CONSTRUCTION (100 FT x 40 FT x 15 FT)

Weather Conditions			Desired	Required	Steam Required Lb Hr	
DB	DP	Percent RH	Percent RH	DB	Heating	Dehumid.
80	73	80	45	98	356	85
70	64	80	45	88	356	72
60	54	80	45	78	356	57

tive humidity; the temperature in the warehouse remains the same as the weather. The difference in operating cost between heating and dehumidifying will offset the additional cost of the dehumidifying equipment for this particular installation in less than two years.

The application of dehumidification to industrial and commercial problems is so broad that it is difficult to cover it in the short period of time that this presentation will allow. In general, the applications may be divided into seven groups or classifications for:

1. Prevention of corrosion and tarnishing of metals, precision tools, instruments, photographic apparatus, and allied equipment.
2. Preservation of the dielectric properties of radio and electronics parts and communication equipment.
3. Prevention of deterioration of foods and food products through the action of dampness and mold.
4. Prevention of damage by moisture to paper, paper products, leather, fur and fur products, textiles, books, and other similar materials.
5. In the production and preservation of chemicals, drugs, pharmaceuticals, and cosmetics.
6. Metallurgical processes.
7. Industrial processing of materials and products.

Dehumidification has become an important part of our industrial and commercial business, and will continue to be, so long as there is moisture in the atmosphere.

It is well to inject a word of caution at this time. Dehumidification is primarily an engineering subject. No credence should be given to the rule-of-thumb methods now being used to analyze this type of problem. Unless the application is properly engineered, the chances of grief are many.

The development of new sorbent materials and the more efficient use of sorbents are making dehumidification an industry comparable to those of heating, refrigeration and air conditioning.

DISCUSSION

W. L. FLEISHER, New York, N. Y.: I want to stress the importance of the careful use of the glycols.

The author and I started this investigation together many years ago, and at that time we were using lithium chloride rather successfully. We attempted to use diethylene glycol and, although we obtained a reduction in the moisture content, the difficulty of using it and eliminating it was so great that, not knowing anything about its toxicity, we dropped it, fortunately.

The glycols have very definitely different vapor pressures, and vaporization of glycol is a function of the vapor pressure. In other words, diethylene glycol will vaporize to produce the same results as triethylene glycol with 10 times as much liquid evaporating. Not only that, but it is toxic and consequently the use of any of the glycols below triethylene glycol is definitely dangerous. You can obtain the same results, but when you make use of evaporation under certain conditions, in order to humidify, you evaporate 10 times as much diethylene glycol as you would with triethylene glycol. The triethylene is non-toxic, and diethylene is toxic.

Also, it is almost impossible, in spraying the glycols, to eliminate the unvaporized droplets by means of the ordinary method, such as eliminators. The droplets have a tendency to float with the air current and, unless they are properly eliminated, they carry through and deposit everywhere at the temperature which may be lower than the dew-point of that particular vapor.

We also have, in some of our so-called haloids, the danger of carry-over which may be toxic.

I do not believe the author mentioned the fact that we had worked with zinc chloride, which was definitely toxic, and which had to be given up.

I am just mentioning these points, and also a point that he might have brought out but did not. It is that some of the advantages of the method, from an economical angle of the use, are that in dehumidifying, a change from latent to sensible heat occurs and your high temperature waters will carry your wet bulb down to a point where you can re-evaporate without refrigerants.

E. G. CARRIER, Boston, Mass.: I assume that the question of preserving materials by dehydration of air, and the setting of a low relative humidity for this purpose, is caused by the necessity of getting the dew-point of the air low enough so that condensation will not take place on materials. We cannot always do this by just reducing relative humidity of air to 45 percent, and I wonder if a great many times we cannot preserve materials better by heating. The only amount of heat needed is that necessary to keep the temperature of the materials up above the dew-point of the surrounding air, and in most parts of the United States, the temperature required is not much above 76 deg. I know that is true in New England. We have had calls for systems for rooms in which steel was stored, and people have requested dehumidification systems to go down to very low moisture contents. The problem has been solved by merely heating the air and not permitting the temperature to go lower than the highest surrounding dew-point. If the material is not colder than the dew-point, no condensation can occur, and if that is the thing against which one is guarding, heating is more economical than dehumidification.

H. M. HART, Chicago, Ill.: Does this include the cost and depreciation of the installation, or just the cost of operation?

MR. EVERETTS: Just operating costs, Mr. Hart.

B. R. SMALL, Pittsburgh, Pa.: This paper is something of a trail blazer in cost studies of dehumidification. The author starts with straight refrigeration, and then simple absorbents, and then he continues with the combination of both. Presumably with the 70 deg effluent air temperature he uses first stage after cooling with water and then refrigeration is the second stage. Perhaps some day the author will continue with the next logical chapter, *the right dehumidifier for specific applications.*

I would like to ask the author whether, in figuring the operating cost of his sorbents, he used the solid adsorbents instead of the liquid? Also, for the amount of reactivation energy, did the author figure about 1 kwh per pound water and in the case of steam, 3 to 4 lb of steam per pound water absorbed?

The author mentions his preference for steam reactivation. Of course, steam must be up around 100 psi gage for solid adsorbents. I have no interest in the gas companies, but on a number of installations, natural gas is working beautifully in order to provide the 300 F reactivation temperature. The water vapor of combustion apparently is not serious. It is something like 4 grains per cubic foot added to the reactivation air.

If you are interested in gas, it just checks out that natural gas at 36 cents per 1000 cu ft is the equivalent of the 30-cent steam cost used by the author.

AUTHOR'S CLOSURE: In line with Mr. Fleisher's discussion, the only reason I did not mention the experiments with zinc chloride is that I still remember the holes it wore in my shoes from the corrosion, and the same thing happened to one or two other materials that we used. But we did have a lot of fun trying to find out just what those things would do.

Regarding Mr. Carrier's comment, the heating of a building, as I did mention, can be used. However, we found in certain applications that the materials which might be stored in that building may not be able to stand the temperature required. I know of one building on the west coast, where in order to maintain 45 percent, we had to obtain a dry bulb temperature of about 112 F, and there are certain materials which you cannot store at that temperature. It is purely a question of application, and the material as well.

In answer to Mr. Small's comment, the cost of adsorbents is fairly well equalized. The amount of material you use in a given system, compared to the total cost of the system, is not too much different. The operating economy, I purposely left out because the next speaker will discuss that, and we have found, on solid adsorbents, that we have gotten down as low as 0.58 kw per pound of water removal as our power input.

I rather feel like a little traitor up here, coming all the way from California where gas is the thing and not having mentioned gas reactivation. Gas is an excellent method of reactivation and is a little better than steam if you do not have available high pressure steam, and out on the west coast and through Texas and those other places where we do use gas we get some very excellent economies from it.

1308

No. 1308

RATING DYNAMIC DEHUMIDIFICATION EQUIPMENT

By Elmer R. Queer* and E. R. McLaughlin,** State College, Pa.

THE United States Navy postwar inactive fleets are presently being put under preservation in what has been popularly called *Operation Zipper*. The chief purpose of the preservation program is economically to maintain the ships in a condition of readiness for 20 years or more in the event another national emergency should necessitate their quick return to service. It is of utmost importance that all equipment and stores except perishable items, ammunition, and gasoline be kept secure in place aboard the ships. Experience following World War I in the *Red Lead Fleet* with heavy grease demonstrated the need for new and more effective means of preventing corrosion, verdigris, tarnish, mildew and rot. All these items are the effect of moisture on materials with which the ship and its equipment are constructed. Dehumidification, or the removal of moisture from the air, is the major new development in this program, and it is the most notable of the various specific methods employed.

In 1924, preliminary work was started by the Navy with low humidity atmospheres aboard inactive submarines, to prevent the electrical circuits from grounding. In this and subsequent work it was determined that if a 30 percent RH atmosphere could be maintained within the vessel, the deleterious effects of excessive moisture would be arrested. Recent experience with test samples of untreated and polished mild steel, suspended for a prolonged period of two years in a dehumidified atmosphere of 30 percent RH, has confirmed previous results regarding corrosion. They have remained bright, and perfectly free of corrosion and dust. Desiccation in these early experiments was obtained with calcium chloride. This proved unsatisfactory because it necessitated entering the ship frequently to remove the solution. In handling, small amounts of the solution were spilled, and excessive corrosion occurred where it came in contact with the ship. Later, automatic machines employing silica gel and activated alumina were used, but the imminence of war placed all available ships in

*Professor of Engineering Research, Engineering Experiment Station, The Pennsylvania State College. Member of A.S.H.V.E.

**Assistant Professor of Engineering Research, Engineering Experiment Station, The Pennsylvania State College.

Presented at the 53rd Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January 1947.

service and made it necessary to delay the preservation project for more urgent work.

Late in 1943, the Bureau of Ships again undertook the task of further developing the technical policies and procuring the necessary equipment to preserve the large number of inactive ships that would be on hand in peace time. The chief problem was to secure a dry atmosphere with the best economy obtainable, and with the minimum of attendance and maintenance. Since temperature control was not considered essential, it was decided to continue with machines employing solid chemical desiccants. The main function of the machines, briefly stated, was automatically to remove water from the ship's air, pass it overboard, and return the dry air to the ship with sufficient pressure to assure adequate distribution of small quantities of air through the fire mains. This dehumidification problem differs from most desiccant drying problems, in that very low and uniform dew points are not required. The equipment need only remove moisture from the ship and be of sufficient capacity to maintain an atmosphere of 30 percent RH.

Several manufacturers built equipment of this kind without incorporating all the characteristics considered necessary, *i.e.*, maximum water adsorption at good power economies, along with reliable operation. These characteristics were a *must* in a program that would be faced with a declining budget and a small attendant force in future years. In order to secure information on equipment being built and to develop specifications, a testing program was instituted at the Engineering Experiment Station, Pennsylvania State College. The climatometer facilities installed early in the war in this laboratory by the Office of Production Research and Development, War Production Board, were considered to be well adapted for work of this nature and they were made available. With these facilities, accurately controlled humidity and temperature conditions could be obtained for rating the dehumidification equipment.

Originally the chief purpose of the testing program, as reported herein, was to develop an adequate test procedure for securing capacity and performance data upon which to formulate specifications for procurement of the equipment. Simultaneously with the capacity tests, performance tests were made on the long-time operation of the machines, to determine the durability of the operating mechanisms and the desiccants. Capacity measurements were made at three-month intervals in the long-term tests. Subsequently, check tests were made on an occasional machine, picked at random from the assembly line, to assure that production continued to meet specifications.

TEST PROCEDURE

The primary objective of the test procedure was to determine the moisture removal rate when air at specified conditions was circulated through the dehumidifier. The moisture which was removed from the air stream was adsorbed by the desiccant and retained until reactivation was effected. Consequently, it was apparent that one method of determining the moisture removal rate was to weigh the desiccant bed before and after the adsorption phase of each cycle. As it was not practicable to weigh the desiccant alone, the entire dehumidifier was mounted on platform scales, as shown in Fig. 1. In some cases the total weight of the dehumidifier and attachments was as much as 1500 lb. The total

weight of moisture adsorbed during a single adsorption period was seldom in excess of 10 lb. This required a reliable mechanism in order to reduce the error of observation. The platform scales were equipped with a precision indicator, which served a dual purpose. It was necessary to have a certain amount of duct work attached to the dehumidifier during test, and the precision indicator assured that the scale platform came back to the same level for each observation and automatically kept the tare weight constant. This indicator also permitted observations to 0.1 lb, which was accepted as sufficiently accurate for determining adsorption weights of 2 lb or more. Before the adsorption was started, known weights were added to the dehumidifier and the scales were balanced. This was used as the *zero* or datum weight for the determinations. As adsorption progressed, and the weight of moisture in the desiccant increased, increment weights were removed from the known weight to keep the scales balanced on the indicator *zero*. The weight removed was equal to the weight of moisture adsorbed. During reactivation, as the weight of moisture in the desiccant decreased, weight was again added to the dehumidifier to keep the scales on *zero*.

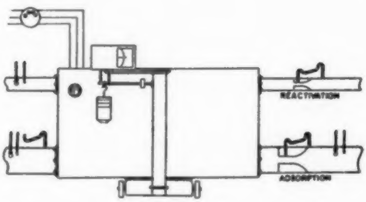


FIG. 1. ARRANGEMENT OF TESTING EQUIPMENT

Frequent and timed observations of the weight permitted plotting a curve of the weight of moisture removed. The slope of this curve is proportional to the rate of water adsorption.

An alternative method for determining the moisture removal rate and checking the weights was to utilize the psychrometric readings obtained from the inlet and outlet air streams, to calculate the weight of moisture adsorbed. This required calibrated equipment and a definite technique for observations. A calibrated orifice was used to meter the air flowing into the unit. The test conditions were held constant and the calculations of the volume and weight of air handled were simplified considerably by metering at the inlet. Inlet air conditions were fixed by maintaining the test room conditions constant. This was readily achieved by placing the dehumidifier and the test equipment in the test room, from which inlet air was taken directly. The outlet air was discharged to the weather, since it varied over a wide range of temperature and dew-point during each cycle. Make-up air leaked in through openings in the wall and mixed constantly with the conditioned air. Variations of the effluent air made determination of the conditions difficult. The dry bulb temperature was seldom constant and the simultaneous wet bulb temperature was difficult to obtain. After considerable checking with a dew-point cup and a humidity analyzer, a wet bulb technique was developed which gave reliable results. A very important part of the technique was timing. In order to have an accurately timed wet bulb

reading, it was necessary to insert the wet bulb thermometer into the effluent stream two minutes before observation time.

In converting the wet and dry bulb readings to dew-points or specific humidities, it was necessary to use the proper barometric pressure. A large-scale psychrometric chart for 29 in. Hg barometric pressure was prepared for this purpose. Specific humidity in pounds of water per pound of dry air was plotted

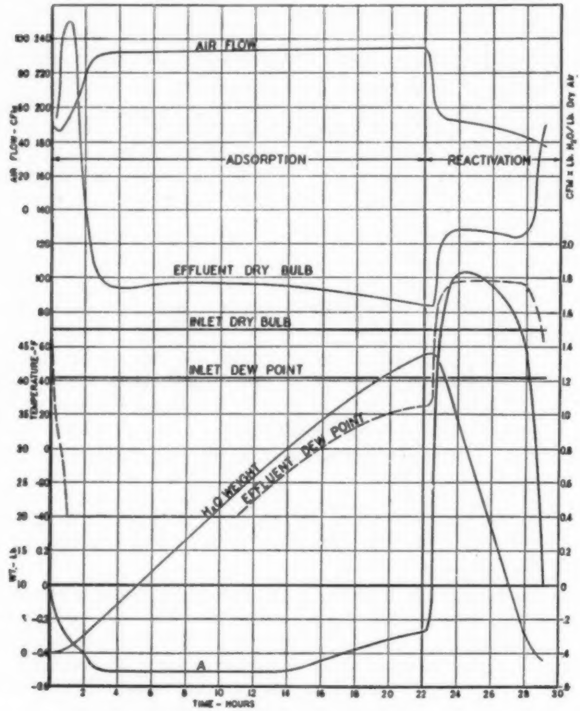


FIG. 2. CHARACTERISTIC CURVES FOR A DEHUMIDIFIER USING A THICK BED OF SILICA GEL

against time. The difference between inlet and effluent humidities represented the moisture removed from each pound of air as it passed through the dehumidifier. This difference multiplied by the instantaneous rate of air circulation gave the instantaneous rate of moisture removal. A summation of the instantaneous rates of moisture removal gave the total weight of moisture adsorbed during the cycle. This weight was compared with the changes in weight observed by the scale readings. When the data for the adsorption and reactivation phases of the cycle are utilized, the weights for the two phases should be the same when the scales indicate that the cycle is balanced. In other words, the water content

of the desiccant should return to its exact original condition. A plot of the calculation of instantaneous rates of moisture removal appears as curve *A* in Fig. 2. A summation of the instantaneous rates of moisture removal was made by measuring the areas between the zero base line of cfm × lb H_2O/lb dry air and curve *A* in Fig. 2. For the adsorption part of the cycle the area is 19.73 units. Each unit of area is equivalent to 2.15 lb of moisture. This gives a total adsorption of 42.5 lb of moisture. For this test the scales indicated a total adsorption of 43.9 lb of moisture. The observed weight was 3.3 percent greater than the calculated weight.

The area under the reactivation phase of the cycle is 20.1 units, representing a total desorption of 43.4 lb of moisture. The weighings indicate that a total of 44.9 lb of moisture were removed by adsorption. (The desiccant was more completely activated at the end of the cycle than it was at the beginning). The observed weight was 3.5 percent greater than the calculated weight.

The percentage difference in each case is within reasonable allowable limits. However, the consistency and the positive signs of these percentages is interesting. Leakage into the system down stream of the metering orifices probably accounts for the discrepancy. On this unit, the inlet and desiccant bed were under negative pressure. Leakage on the effluent side of the fan would have very little effect on the calculations.

SINGLE-BED DEHUMIDIFIER

Fig. 2 shows the characteristic curves for a single-bed silica gel type of unit operating with parallel flow, *i.e.,* the reactivation air flows in the same direction through the desiccant as the air for adsorption.

A consideration of the individual curves will illustrate the performance of a dynamic unit. At the beginning of the cycle, the bed was *activated.* A large portion of the desiccant bed was heated to a temperature of 250 F or higher. When the effluent temperature reached 190 F, a thermostat actuated the valve mechanism and the air flow for adsorption was established. It was necessary for the air to cool a layer of the bed before adsorption could take place. As the heat in the bed was transferred to the air stream, the effluent air was very hot for the first part of the cycle. In addition to the residual heat in the bed, an energy exchange was taking place in the zone of adsorption, where the heat of adsorption was also transferred to the air stream. This accounted for the consistent temperature rise in the dry bulb temperature of the air as it passed through the unit during the 8 hr period, 3 hr to 11 hr. During this time the rate of adsorption was constant (see Fig. 2, curve *A*). The temperature rise of 26 F represented a moisture removal of 39 grains per pound of air or 2.8 grains per cubic foot of entering air. This is approximately 9 deg temperature rise per grain removed per cubic foot. After 11 hr, when the *break point* in the effluent dew-point curve was reached, the adsorption rate decreased and the outlet dry bulb temperature decreased as adsorption continued. The adsorption phase of the cycle was controlled by a timer which in this instance stopped adsorption at 22 hr.

The outlet dew-point curve indicated that adsorption started as soon as a thin layer of the desiccant was cooled sufficiently to permit adsorption. A very thin layer of the desiccant cooled quickly, and adsorption began almost immediately. However, it required some time before the cooled portion was thick enough to

reduce the dew-point below —30 F. In Fig. 2 this time is shown to be approximately one hour. During the next several hours, or until 11 hr had elapsed, the effluent dew-point remained low, approximately —40 F and a high percentage of the moisture in the entering air was adsorbed. Following the break point, the effluent dew-point curve rose very rapidly and would ultimately have become asymptotic to the inlet dew-point curve if the timer had not interrupted adsorption at 22 hr.

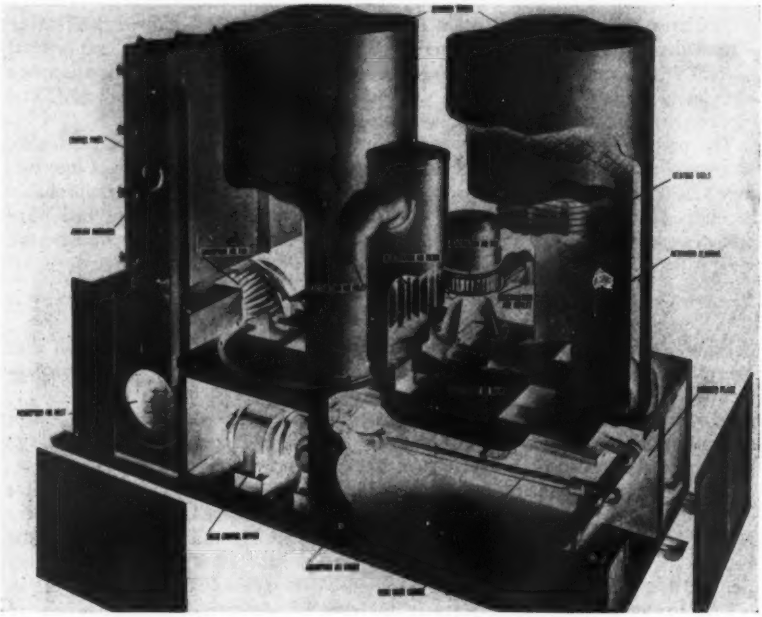


FIG. 3. A TYPICAL DUAL-BED DESICCANT DEHUMIDIFIER

The observed weights are plotted in Fig. 2, showing how the adsorption weight increased at a constant rate until the break point was reached and the rate gradually decreased. At 70 F and 35 percent RH, the quantity of desiccant used in this unit has a potential adsorption capacity of approximately 50 lb of moisture. In order to maintain a satisfactory average moisture removal rate, it was necessary to terminate adsorption before the ultimate capacity was reached; however, not before the break point was reached. Another consideration was the fact that the nearer saturation is approached during adsorption, the more economical will be the reactivation.

For the first half hour after the reactivation circuit was established, the net result of the operation of the unit was a slight adsorption. This lag was caused by the heat capacity of the bed. After ½ hr heating, the reactivation started and

the rate increased very rapidly. The outlet dry bulb (Fig. 2) increased to a peak and slowly fell off until reactivation was nearly complete. This phenomenon of falling dry bulb temperature is typical of units incorporating parallel flow, *i.e.*, the reactivation air flows through the bed in the same direction as the adsorption air. When reactivation began, the inlet face of the bed was nearly

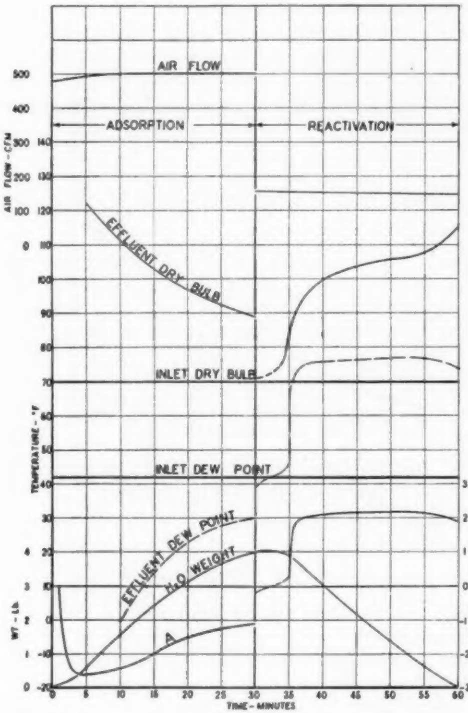


FIG. 4. CHARACTERISTIC CURVES FOR A DEHUMIDIFIER USING A THIN BED OF ACTIVATED ALUMINA

saturated. Consequently, the moisture removed from the desiccant near the inlet was carried through the bed and part of it was readsorbed by the partially saturated desiccant at the outlet. The heat released during readsorption raised the temperature of the air above the normal reactivation effluent temperature. As reactivation progressed down through the bed, the readsorption diminished and, just before reactivation was completed, it ceased altogether. This decreasing readsorption accounted for the decreasing dry bulb temperature as reactivation progressed. When the desiccant was completely reactivated, heat for de-

sorption was no longer required and the outlet dry bulb rose very rapidly to approach the temperature of the air leaving the heaters and entering the bed. This rapid rise in temperature was used to terminate the reactivation phase of the cycle.

These curves illustrate the performance of a dynamic dehumidifier. A single bed unit having a thick bed of desiccant was chosen for analysis, to provide a long cycle incorporating all the features of a dynamic unit.

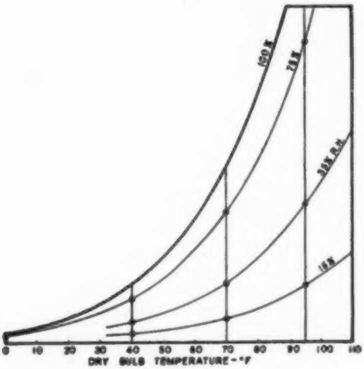


FIG. 5. DISTRIBUTION OF TEST POINTS ON PSYCHROMETRIC CHART

DUAL BED DEHUMIDIFIER

A unit employing two desiccant towers and a short cycle is more adaptable to the ship preservation program. Units of this type (Fig. 3) circulate a larger volume of air through thin beds, one bed being reactivated while the alternate bed is adsorbing. These features change the characteristic curves materially, and the test procedure had to be altered to provide the required data. In order to obtain scale weights, it was necessary to make one bed inoperative while observations were being taken on the operating bed. The cycles were short, ranging from 30 min to 3 hr, and it was very important to time the observations properly. The equipment shown in Fig. 1 was used for the tests on dual-bed units. Fig. 4 represents curves observed on a unit of this type employing activated alumina as the desiccant. It will be noted that there is some similarity to the curves in Fig. 2. The bed is thin, and the air velocity high enough to eliminate the break point completely. A timer controls the cycles and the bed is not reactivated to dryness. These units employ counter flow during reactivation, and readsorption characteristics do not appear.

The area under the adsorption rate curve A in Fig. 4 is 10.9 units. Each unit is equivalent to 0.36 lb of water, and the total adsorption is 3.93 lb of water which compares with 4.0 lb indicated by the scales.

The area under the reactivation rate curve is 10.5 units, representing a weight of 3.8 lb of water. This is 95 percent of the observed weight of 4.0 lb of water. The discrepancy of 5 percent is logically attributable to leakage into the reactivation circuit, since the air was under a slight negative pressure as it flowed through the dehumidifier. A test for leakage disclosed 5 cfm when the circuit was subjected to a pressure differential of 0.55 in. water.

The curves in Fig. 4 are typical of a dual-bed unit meeting the requirements for shipboard use. Total adsorption is not attempted nor is it necessary. Only the moisture in excess of 35 percent RH is conducive to corrosion, and it is more desirable to reduce the humidity than to remove it totally, since the nearer total adsorption is approached, the more expensive the operation becomes.

A dual-bed unit also provides a continuous source of dehumidified air and has

TABLE 1. PERFORMANCE DATA OF DEHUMIDIFIERS

Inlet Air Condition			Activated Alumina Machine			Silica Gel Machine			Silica Gel Machine		
DB F	DP F	RH %	H_2O Ads Rate lb/Hr	Economy kwhr/lb H_2O	Machine Data	H_2O Ads Rate lb/Hr	Economy kwhr/lb H_2O	Machine Data	H_2O Ads Rate lb/Hr	Economy kwhr/lb H_2O	Machine Data
95	64	35	8.6	0.83	500 cfm-Ads	9.4	0.70	430 cfm-Ads	7.5	0.84	400 cfm-Ads
70	42	35	8.0	0.92	5.5 in. SP Ads[a]	7.6	0.88	5.5 in. SP	7.2	0.92	5.5 in. SP
40	15	35	3.8	0.96	150 cfm-React	3.1	1.10	156 cfm-React	3.3	0.92	79 cfm-React
0	−6	75	1.8	2.06		1.2	6.58		1.5	1.50	
95	64	35	2.9	0.77	150 cfm-Ads	4.7	0.80	250 cfm-Ads	5.8 lb/day	0.82	20 cfm-Ads
70	42	35	2.4	0.91	3.4 in. SP Ads	4.0	0.93	5.5 in. SP	5.5 lb/day	0.85	⅛ in. SP
40	15	35	1.3	1.81	70 cfm-React	1.8	1.25	82 cfm-React	3.2 lb/day	1.53	15 cfm-React
0	−6	75	0.5	5.37		0.8	2.96		1.2 lb/day	4.68	

a Static Pressure — Adsorption Outlet.

a greater capacity per pound of dehumidification equipment. By the use of thin beds and other features, the dual-bed unit can operate at maximum economy.

The economic characteristics of the units are readily compared by the power costs for reactivation. The economy is expressed as kilowatt-hours per pound of water desorbed; consequently, a low figure represents an efficient operation. It was desirable that the economy should not exceed 1 kwhr per pound. After analysis of the initial test results, the manufacturers were able to incorporate new features in their designs to reduce the economy for a large part of the operation considerably below 1 kwhr per pound.

TEST CONDITIONS

In addition to developing a test procedure, it was necessary to establish the conditions at which tests were to be made. Specific conditions of temperature and humidity were chosen to facilitate comparisons of the performance of individual units. For this purpose, the 10 points represented in Fig. 5 were selected. These test conditions do not include the extreme conditions under

which a dehumidifier may be required to work, but they do cover the normal field of the psychrometric chart.

For effective preservation, the ship's space is to be maintained at approximately 30 percent RH. For that reason the 35 percent RH line was chosen and tests at three temperatures, 40, 70 and 95 F were selected to provide data on the effect of temperature on performance. During the initial drying period after installation aboard ship, the dehumidifier will operate on air of relatively high humidity. For the purpose of determining performance under this condition, the 75 percent RH tests were chosen. To provide a third point at each temperature, the 15 percent RH tests were chosen.

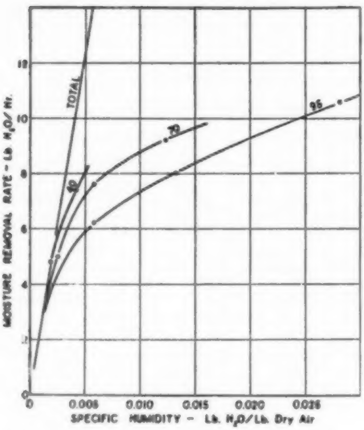


Fig. 6. Capacity Curves for Dual-Bed Dehumidifier

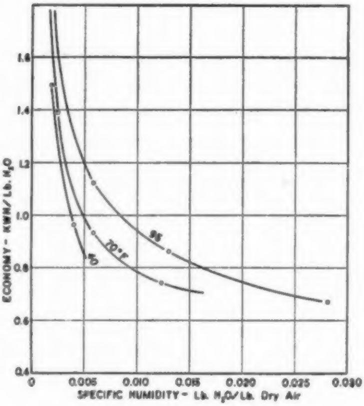


Fig. 7. Economy Curves for Dual-Bed Dehumidifier

Of these tests, the most attention was focused on the 70 F, 35 percent RH test, since it was the most representative. Considerable testing was done under the other two 35 percent RH conditions. A test at 0 F was made, primarily for observation of the mechanical operation at low temperatures and, although an attempt was made to determine capacity, it was difficult to observe weights accurately at this low temperature.

Performance Summary

A summary of the performance of typical units is given in Table 1. These data are for the four basic test conditions. Fig. 6 is a plot of the data for a dual-bed unit and indicates the effect of temperature and specific humidity of the air entering the unit upon capacity. The curves indicate that, for a given entering moisture content, increase of the dry bulb temperature will reduce the performance of the dehumidifier. The power requirement of reactivation under these same conditions is shown in the plot of economy vs. entering humidity in Fig. 7.

Units effecting dehumidification by means other than solid adsorbents were also tested with only slight modifications in the test procedure.

The test procedures and methods described herein were developed for a specialized problem. Few industrial problems will compare with the ship preservation program. Consequently, modified dehumidification equipment will be necessary for each type of application. Modifications can be guided only by experience or research. Perhaps the most important part of the Navy program, next to preserving the fleet, was the initiation of a test program to guide the services and industry in the specification and selection of dehumidification equipment. As interest in the control of moisture and the adverse effects of moisture on materials increases, the requirements for dehumidification are going to increase. It is hardly necessary to say that the problems will increase. It is hoped that this paper may serve as a nucleus for the development of test procedures for other dynamic dehumidification units.

ACKNOWLEDGMENT

The Pennsylvania State College Engineering Experiment Station wishes to acknowledge the fine spirit exhibited by the Office of Production Research and Development of the War Production Board in withholding their testing program and making the Climatometer facilities they installed in this laboratory available for the Bureau of Ships dehumidification research program. The authors are indebted to Prof. F. G. Hechler, director of the Engineering Experiment Station, for the valuable counsel given during the program; and the assistance rendered by L. E. Epstein, Bureau of Ships, in developing the test procedures and securing test data. The release by the U. S. Navy Bureau of Ships of the data for publication is duly acknowledged.

DISCUSSION

T. H. URDAHL, Washington, D. C.: I wish first to congratulate the authors on a very clear and understandable paper, and comment that this very practical means of testing not only provided necessary information as to the capacities and performance of dehumidification equipment, but that it was also the means whereby the proper materials of construction and tolerances to be required in the construction of machines were established.

To those who have the manufacturing problem, this matter of close tolerances often involves a considerable headache, especially when machines must be produced en masse. A single machine can be made to produce a result, but to build many machines that will identically produce a consistent result, calls for manufacturing tolerances that are often lost in production.

The testing program that Professor Queer outlined to you was the Navy's means of checking the performance of machines as they were delivered, and so far service results have demonstrated that the construction employed in the machines will give the long life which he mentioned at the beginning of the program, and which the Navy desired, a twenty-year equipment life, with a minimum of repair.

P. J. MARSCHALL, Chicago, Ill.: I have been extremely interested in this paper and Mr. Everetts' paper. However, I think the application of dehumidification to an industry such as that with which I am associated, which happens to be the drug, chemical and pharmaceutical industry, presents more problems than found in the case of ships.

We have to maintain low relative humidities in rooms in which highly hygroscopic powders are processed. The humidity must be maintained at less than 15 percent, with a dry bulb temperature at approximately 80 F.

We have a number of dehumidification systems employing both types of sorbents, solid and liquid. Our experience indicates that contamination of the sorbents, with either solvent vapors or dust from the product being processed, affects the efficiency of the dehumidifier.

It is essential to employ highly efficient dust collecting equipment ahead of the dehumidifiers to insure performance in accordance with the ratings you have seen on the charts.

Obviously, it is also unsatisfactory to use gas for reactivation if there is any possibility of contamination of recirculated air as a result of the use of solvents within the area.

AUTHORS' CLOSURE: Mr. Marschall has touched on a very important point, regarding the contamination of desiccants. Recently we had an occasion to contaminate some desiccants in order to determine the machine performances under this condition. On one particular type of desiccant we used fuel oil in atomized particles, turpentine, lubricating oil and dust as well as heavy black smoke from burning oil and rubber. Under certain conditions the adsorptive capacity improved slightly. At the manufacturer's suggestion we tried powdered sugar. Either we did not use a sufficient quantity or sugar likewise had little effect on machine capacity.

In the Navy program a little difficulty was encountered with an asphaltic residue accumulating in certain types of desiccants. The effect of this contamination was to reduce the adsorptive capacity of the desiccant involved by as much as 50 percent in some cases. The cause of this has not been definitely determined. It is believed in the general development of desiccants the dynamic characteristics will be still further improved, so that the effects of contaminations will be minimized.

1309

No. 1309

METHODS USED IN DETERMINING THE HEALTH HAZARDS ARISING FROM THE INHALATION OF VARIOUS CHEMICALS

By FRANCIS F. HEYROTH,* M.D., CINCINNATI, OHIO

INTRODUCTION

KNOWING the volume of air in a given locality and the rate at which it is being contaminated by a given mist, dust or vapor, a ventilating engineer could estimate the rate of ventilation necessary to lessen the contamination to any desired limit. To set the limit which must not be exceeded, if the area is to be made safe, obviously requires a knowledge of the toxicity of the contaminant. It is the author's intention to acquaint ventilating engineers with the methods available for measuring the toxicity of air-borne contaminants, the reliability of the data secured by the various methods most frequently employed, the manner in which agreement may be reached as to the maximum allowable concentration of a given substance, and the sources of information that are available.

BRIEF HISTORICAL REVIEW OF THE DEVELOPMENT OF QUANTITATIVE INDUSTRIAL TOXICOLOGY

Prior to the past 30 to 40 years, few if any attempts were made to learn anything of the relationship between the incidence of industrial intoxication, acute or chronic, and the degree of exposure to the toxic agent to which workmen were subjected. Not until shortly after 1900 was it established that the chief agent in the causation of lead poisoning was lead-bearing dust or fume suspended in the air, and that the lung was the chief portal through which lead was absorbed. This knowledge gave rise to efforts to learn the amount and particle-size distribution of dust present in the air surrounding certain processes known to be especially dangerous, and to corrrelate these with the incidence of industrial poisoning. About this time, a few rather crude experiments were devised in

*Kettering Laboratory of Applied Physiology, College of Medicine, University of Cincinnati.

Presented at the 53rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January 1947.

which, for an hour on each of a number of successive days, animals were forced to breathe air contaminated to various degrees with lead dusts.

Although much effort was devoted during the next 25 years to the collection of statistics regarding the incidence of illness from exposures to various other toxic substances in industrial use, and to the clinical description of the intoxication, quantitative data relating incidence of illness to the severity of exposure which induced it were but rarely sought. In 1912, a European text on industrial poisoning made mention of such data derived from animal experimentation, but only in the case of benzene, carbon disulfide, chlorine and ammonia. As late as 1925, the comprehensive book by Alice Hamilton, which did much to stimulate interest in industrial intoxication, provided figures regarding the atmospheric concentrations required to induce varying degrees of intoxication in the case of fewer than 10 substances.

The importance of securing quantitative data depends upon the recently appreciated fact that any toxic substance may be absorbed up to a certain amount without inducing detectable harm. The margin between this limit and the amounts required to induce severe or fatal injury varies widely with the nature of the toxic agent.

Animal experimentation offers the only means of studying at will the effects induced by the inhalation of air bearing a toxic agent in a variety of predetermined concentrations. It offers, however, many technical difficulties, which were only gradually surmounted. Beginning about 1880, Lehmann, a German hygienist, and his students, began such investigations, studying at first the halogens, hydrogen sulfide, carbon disulfide, and a few other gases. After 1900, work of this nature was continued at Würzburg, both by Lehmann and by Flury and Zernik. Sayers and others soon began such investigations at the U. S. Bureau of Mines. The industrial developments arising from the first World War stimulated others to enter this field, and experience in the study of war gases led to theoretical and technical advances in the methods employed. For the most part, however, this work was limited to a study of the immediate and usually severe poisoning that results when air bearing such gases in known but relatively high concentrations is inhaled over a period of from a few minutes to an hour or so. In a relatively few instances, an effort was made to learn the maximum concentration that could be borne safely for six hours.

It is now realized that such brief experiments are quite incapable of revealing any useful information relative to the concentrations which give rise to chronic intoxication, such, for example, as might result from the daily handling of certain materials in industry. One of the first investigations, in which it was sought to learn the maximum concentration in which a gas could be tolerated in the air inhaled throughout the entire working day, and on every day, was conducted by Henderson and associates, 1921-22, upon carbon monoxide, in order to learn the ventilation necessary in the Holland tunnel. At present, it is the goal of the industrial toxicologist to supply information of use in setting a standard for the concentration of a toxic substance which should never be exceeded in the air of rooms in which men and women work. For many substances, two or more standards are needed: one so low that no harm can result even when the period of exposure extends over every working day in the year, and the other, a higher one applying only to shorter periods of exposure, such as may result from intermittent operation or from leaks in equipment. Much work

on both immediate and chronic toxicity has been and is being done in university laboratories (notably at Chicago, Cincinnati and Harvard); in the laboratories of the Division of Industrial Hygiene, *U. S. Public Health Service;* at the Mellon Institute; and elsewhere. In large part, the work is directly supported either by industry or from governmental sources. At present, data exist regarding the toxicities of well over 100 gases, vapors, dusts, fumes, and mists. These, however, are not of uniform reliability. At present, the trend is being established that no new compound shall be handled until its toxicologic properties have been investigated, at least to some extent.

METHODS USED IN THE QUANTITATIVE EVALUATION OF TOXICITY OF AIR-BORNE MATERIALS

In general, there are three sources of data necessary for setting maximum allowable concentrations: field observations, experiments upon animals, and experiments upon man.

Field Observations

The first of these was long the only one available. It involves the correlation of the results of frequent medical examinations of the employees with data secured by analyses of the air in which they work. The use of this method has a number of serious limitations. It requires the cooperation of both chemists and physicians, each of whom must be specially trained for this work, and capable of exercising a high degree of critical judgment. The chemist may have to devise methods, sometimes microchemical, for the determination of very small quantities of a new organic substance in the air. The techniques for the enumeration of dust particles and for determining their size distribution are unfamiliar to most chemists. But it is in determining how often and where to collect the samples that the greatest judgment is needed. This requires a thorough knowledge of how and when the process is carried out, and the vagaries in it that occur from time to time, as well as of the numbers of men employed, their location with respect to the equipment, and the degree of severity of their labor. The physician must also know these facts in regard to the operating conditions if he is to distinguish the illness he may encounter from that arising from causes less intimately associated with the process in question. When the maximum allowable concentration which it is sought to establish is but slightly exceeded, the symptoms of intoxication are apt to be of a mild and indefinite nature, which may give rise to confusion with illness or lowered efficiency arising from such causes as poor nutrition, fatigue, unsanitary working conditions, etc. He must therefore know thoroughly the usual state of health of men engaged in similar operations, but not exposed to the toxic agent. In the absence of clues yielded by the other approaches (animal or human experimentation), the physician must be able to evaluate the symptoms and detect the first of the more severe and possibly characteristic signs of intoxication that may actually be attributed to the chance incidence of an excessive concentration of the substance in the air. It is obvious that the maximum allowable concentration can be determined by this procedure alone only in the event that actual intoxication has resulted on one or more occasions. One goal of modern industrial toxicology—the safe introduction of new products into industry—can obviously not be met by relying solely upon this

approach. On the other hand, experience has demonstrated that the method has great value when used in conjunction with other methods involving the experimental approach.

Animal Experimentation

This offers the advantage that the new toxic agent may be studied in a large range of accurately known concentrations at will, and that the experiments may be conducted for any desired period of time from minutes to months. It also permits the demonstration of the nature of the damage which results from the absorption of the new compound. Observation of the behavior of animals during or after the period of exposure may be of great value in making clear the mechanism by which the agent acts. Examination of the tissues of animals that die during exposure, or are killed after they have survived for various periods thereafter, particularly if supplemented by later microscopic examination, will often reveal which organs are most severely damaged, and show much regarding the nature of the processes by which the damage occurs. This information is of value in disclosing the type of signs and symptoms that should be looked for in the case of workmen. The knowledge that the red cells of the blood are damaged when animals are exposed to vapors of aniline shows that careful and frequent examinations should be made of the blood of workmen who handle this material. The finding of severe damage to the liver in animals following the inhalation of vapors of carbon tetrachloride suggests that those who work with this substance may be protected by subjecting them to tests designed to indicate an impairment of the function of that organ. Furthermore, animal experiments may afford a basis for the recommendation of prophylactic measures, since they have demonstrated that, when adequate quantities of carbohydrates and certain amino-acids are supplied, the susceptibility of the liver to damage by some of the chlorinated hydrocarbons is lessened. This information is of direct value to the industrial physician responsible for the care of such workers, for it enables him to exclude from exposure those poorly nourished persons most prone to suffer injury, and induces him to set up means for prophylactic instruction in nutrition. Finally, a knowledge of the mechanism by which the compound induces intoxication is of value in affording a rational basis for the treatment of such cases of poisoning as may occur through accident or inadvertence.

The use of animal experimentation in setting a maximum allowable concentration has, however, certain limitations, which are inherent in the variations in the response of individuals or different species of animals to exposure. When animals of several species are exposed simultaneously to air bearing a new substance in known concentration, it frequently occurs that those of one species may survive while all those of another may die. Such experiments may therefore yield not one but a series of maximum allowable concentrations, each applicable to a different species. For example, 8 of 20 mice died during a series of 20 seven-hour exposures to a stream of air bearing DDT in the dust in a concentration equivalent to 0.008 mg of DDT per liter, but guinea pigs, rabbits and cats tolerated exposures to much higher concentrations.

Wide variations in the susceptibility of animals to the inhalation of air bearing ketene gas in low concentrations occur. Mice die during or shortly after a 10 min exposure to air bearing 25 to 50 ppm by volume, but greater amounts must be present in the air to cause the death of animals of other species, the lethal

concentrations being 50 to 200 ppm for monkeys, 250 to 375 ppm for rats, 375 to 500 ppm for guinea pigs, and 750 to 1000 ppm for rabbits.

Such variations in susceptibility make it necessary to use great caution in applying to man the quantitative data secured from the response of animals. Qualitative as well as quantitative differences have also been observed in the response of animals of various species. The inhalation of the vapor of monomethylaniline induces in rabbits a rather severe anemia, while cats do not exhibit this response but show instead, a remarkable alteration of the appearance of the red cells, accompanied by the formation of methemoglobin. From these results alone, the nature of the response of man could not be predicted. It is not possible to select in advance, for experimental work, any one species whose response will most closely resemble that of man. Rules based upon assumed similarities, either anatomical, biochemical or phylogenetic are of little general validity. The best that can be done at present is to employ as wide a variety of species as is feasible. In practice, rabbits, rats, mice, guinea pigs, cats, dogs and monkeys are those most commonly used.

Quantitative variation within a given species occurs among both men and experimental animals. Sex, age, genetic strain, nutritional status and the effects of the ordinary diseases that occur among animals are among the more obvious causes of such variation. It has recently been demonstrated that very young rats are at least 200 to 400 times more resistant to the administration of a given dosage of thiourea than are adults.[1] However, even when these factors are as adequately controlled as possible in the design of an experiment, there still remains a considerable variation in individual susceptibility. The capacity of an exposure chamber of a size suitable for laboratory experimentation necessarily limits the number of animals which can be used in a given experiment. Since, in determining maximum allowable concentrations, the animals must be exposed for six to eight hours daily, on at least five days of each week, over a period of six months or more, replications are limited. There can be little assurance that the six to ten rabbits chosen for use in such an experiment represent a truly random sample of all rabbits available, so that the results apply strictly only to those actually used rather than to rabbits in general. The necessity for dealing with such limited numbers of observations has given rise in recent years to the development of new statistical methods that make proper allowance for such limitations.

Quantitative data may also be secured concerning the amounts of the substance that have been absorbed into the blood and tissues of the animals, when methods for its determination can be devised. Analyses of the excreta are also of value in determining the rate at which it is eliminated from the body. Many substances undergo chemical alteration during their passage through the body, and it is frequently of importance to learn the form in which they are excreted. Others are broken down to simpler products or are burned to carbon dioxide and oxygen. Accordingly, efforts are made to determine as completely as possible the fate of the substance while in the body.

Human Experimentation

Human experimentation may be desirable after a sufficient background of information has been obtained from animal experiments to make it safe. It is

---

[1]Dieke and Richter, *Journal of Pharmacology*, *83*:195 (1945).

rarely feasible to employ this method to determine a maximum allowable concentration, since this would necessitate the daily exposure of several men over prolonged periods. It is of value as a check upon the possibility that the value arrived at on the basis of animal experimentation may be grossly in error when applied to man. A concentration somewhat higher than that believed allowable for prolonged exposure may be chosen, and several men be exposed to it for periods up to two or three days. Large chambers in which certain forms of activity may be carried out have been used. Before the actual exposure, the men are given a thorough physical examination, including numerous tests of organic function and a battery of psychological tests. The examination is repeated following the exposure, and any differences that may be observed are noted. Certain of the tests, especially neurological and psychological, may also be given during the actual exposure. The excreta are examined, in order to learn the extent to which the substance may be eliminated from the body, either as such or in altered form.

Experiments of this type are of especial value in appraising the hazards of exposure, in poorly ventilated quarters and for relatively brief periods, to concentrations somewhat in excess of that believed permissible for prolonged exposure. Such experiments have been recently carried out in the study of toluene, insecticidal dusts, and certain fluorinated compounds. In the case of toluene, for example, experiments have shown that the substance, when inhaled in concentrations not injurious to bodily health, nevertheless induced a transient and slight impairment of psychic function which might interfere with the performance of certain operations, or cause lessening of judgment which under certain circumstances might increase the incidence of accidents.

THE SETTING OF MAXIMUM ALLOWABLE CONCENTRATIONS

It is natural in publishing the results of animal or human experimental work, that an investigator should draw such inferences as he believes justified as to the maximum concentration allowable for prolonged exposure. Frequently, however, the values that have been proposed by various investigators for a given substance differ considerably and the work of each must be examined critically before a choice can be made among them. If the compound is in actual use, the results of field correlations between actual exposures and the incidence of illness may also be available for comparison.

In recent years, several groups of workers in the field of industrial hygiene have sought to collect, appraise and correlate the clinical and experimental data that are available in regard to numerous substances. The industrial hygiene units in various state departments of health or labor have been particularly active in this field. The first list of allowable concentrations prepared in this manner, after consultation with many authorities in the United States and elsewhere, was that promulgated by the Massachusetts Department of Labor and Industries in 1937, which was amended and extended in 1939 and 1940. It is still carefully stated to be unofficial and intended only as a guide for the use of the Division of Occupational Hygiene in its supervisory work. This is an essential point, because it takes cognizance of the limitations of both experimental work and actual field operations. It makes clear also that the proposed standards are tentative, pending the accumulation of more data from practical human experience or from more extensive experimentation. Other official groups in New York and Utah,

as well as the *U. S. Public Health Service,* have subsequently prepared published lists of allowable concentrations as guides for their own or other industrial hygienists. On the other hand, some states (California, Connecticut, Oregon and Ohio) have incorporated their own selections of such values into regulations or codes which have the effect of law. Unfortunately, the values set forth by these various agencies do not always agree, and it is possible for a degree of atmospheric contamination that is regarded as dangerous in one state to be held safe in another. Thus, if an industrial organization were to carry on a process in similarly ventilated plants in two states, it might be subjected to legal penalty in one state for maintaining conditions that would be acceptable in the other.

TABLE 1. MAXIMUM ALLOWABLE CONCENTRATIONS
American Standards Association

	PARTS PER MILLION
Arsenic	(0.15[a])
Benzene (Benzol)	100
Cadmium	(0.1[a])
Carbon disulfide	20
Carbon monoxide	100
Chromic acid	0.1[a]
Formaldehyde	10
Hydrogen sulfide	20
Lead	0.15[a]
Manganese	(6[a])
Mercury	0.1[a]
Methanol (Methyl alcohol)	200
Nitrogen oxides	25
Styrene monomer	(400)
Toluene (Toluol)	200
Trichloroethylene	200
X-ray	(0.1r)
Xylene (Xylol)	(200)

a Milligrams per cubic meter
( ) War Standards

For example, the maximum concentration of methyl bromide regarded as permissible in California (500 ppm) is far greater than that (20 ppm) allowed in Ohio. Discrepancies between the standards embodied in legal codes or in unofficial guides of the various state departments of industrial hygiene exist in the case of at least 28 toxic substances.

This difficulty might be minimized if the authorities were to agree to adopt only such values as had been promulgated by some nationally recognized organization such as the industry-supported *American Standards Association,* which has a mechanism by which the pertinent data may be evaluated in an unbiased fashion through a committee of experts representing industry, insurance companies and governmental agencies. An objection that has been advanced against this procedure is that its cumbersome nature would tend to lessen the frequency with which the values might be reconsidered as new data become available. This may, however, be overweighed by the likelihood that such committees, representing as they do a wide variety of expert opinion, will be less likely to propose standards based on only a portion of the available evidence than would a

more narrowly constituted local body. Since 1940, 17 such standards for dusts, gases and vapors, as well as one for X-radiation, have been adopted by the *American Standards Association*, as shown in Table 1. A few of these were adopted tentatively as war standards, later revision being contemplated. In the case of dusts, the units are milligrams per cubic meter of air, but gas or vapor concentrations are usually expressed as parts by volume per million of air. Standards set by other authorities are sometimes stated in terms of weight of gas or vapor per unit volume of air, as milligrams per liter. Tables for the interconversion of these values are available, the table being entered at the molecular weight of the gas. With the exception of not more than three states, the limits set by the *American Standards Association* are those generally adopted in state codes.

It must be emphasized that the adoption of these or other standards does not lessen the need for careful medical supervision in all plants using dangerous chemicals. Records of actual concentrations and their fluctuations should be kept and compared with the results of frequent clinical examinations, and in some instances with data on the concentrations of the substances in the blood or excreta of the workers, or with the results of special tests for alterations of metabolism or physiological function. In some cases, ventilation and other mechanical means for the control of atmospheric contamination may be too costly or difficult, and it may be desirable to change the process in order to use a less toxic substance.

SOURCES OF INFORMATION

The engineer desirous of learning the values adopted officially or semi-officially by the various states, will find them tabulated, as of November 1945, in a recent article.[2] This tabulation does not include the Ohio code, adopted in 1946, but does include those adopted by the *American Standards Association* as well as values which Cook tentatively recommended for a wide variety of materials for which neither the *American Standards Association* nor any state authority had previously proposed a standard. In any event, the engineer responsible for planning ventilation requirements would do well to consult the local state authorities to learn the requirement for the material in question. More detailed information may be sought in text or reference books.[3, 4, 5]

Anyone desirous of going to the original sources will find the earlier German work in the Zeitschrift für Hygiene, while much of the work done in this country since 1919 has appeared in the *Journal of Industrial Hygiene and Toxicology*, or in the *U. S. Public Health Reports* and in special bulletins issued by the Bureau of Mines or the Division of Industrial Hygiene of the *U. S. Public Health Service*.

DISCUSSION

ALLEN D. BRANDT, Bethlehem, Pa. (WRITTEN): Dr. Heyroth's paper brings much information to the attention of the ventilating and plant engineer which should prove of value to him. It should be of particular interest to those ventilating engineers who specialize in exhaust ventilation for industrial atmospheric sanitation.

---

[2]Warren A. Cook, *Industrial Medicine*, *14*:936 (1945).
[3]Noxious Gases and the Principles of Respiration Influencing Their Action, by Henderson and Haggard. (Reinhold Publishing Corp., Second Edition, 1943.)
[4]Flury and Zernik, Schädliche Gase (Springer, Berlin, 1931).
[5]The Analytical Chemistry of Industrial Poisons, Hazards and Solvents, by Jacobs. (Interscience Publishers, Inc., New York, 1941.)

The author was careful to point out that setting ***maximum allowable concentrations*** (MAC), as a rule, requires toxicological data obtained from the three avenues of investigation. This is a very important point, and explains why the engineer sometimes must ***grope in the dark***. He is called upon to design a ventilating system to control the dust, fume or gas liberated in a new operation or process, or in a renovated one where new materials are being used. If perchance he is told what is in the new material, rather than handed some totally non-descriptive trade-name, he is fortunate indeed, for then he can get from publications and tables such as Dr. Heyroth describes, some idea of its harmfulness. What the ventilating engineer desires and frequently needs, however, is concrete data on the maximum allowable concentration. Owing to the time and effort required to establish these concentrations, only relatively few have ever been established and the engineer must resort to ***guestimates*** on the basis of comparative chemistry and toxicology. These values, if based on animal experimentation only, may be erroneous, as the author states. Nevertheless any value, even if wrong, is better than no value at all to the engineer who must design the control measures. This is not nearly so important for those operations which can be enclosed, or to which local exhaust can be applied effectively, as it is for those operations in which general or dilution ventilation plays a significant role, as it does so frequently when one is dealing with volatile liquids.

There is considerable difference of opinion in the industrial hygiene field on this matter of estimating tentative safe limits for new chemicals on the basis of animal experimentation. Many toxicologists are opposed, and not without reason, to guessing at the safe limit for industrial workers from their animal research. They may be wrong and this might reflect discredit upon them. True though this may be, are they not in a much better position to make a guess than the engineer who has no information at all and who does not even have the advantage of a good chemical and toxicological training? The engineer does not want to make a guess, but, if the toxicologist refuses to do so, there is no ***out*** for him unless we are willing to see how dangerous a new chemical is by its effect upon the workers before control is provided. Certainly the engineer wants more MAC's, many more, but in lieu of such values which can be accumulated at only a very slow rate, he will be most thankful for any ***guestimates*** which the toxicologist may provide regarding the comparative harmfulness of the newer chemicals.

DR. WILLARD MACHLE, New York, N. Y. (WRITTEN): I think Dr. Heyroth has done an excellent task in defining what one searches for in toxicologic investigation, and the means by which one obtains an answer. It is evident that the methods apply without regard to the route of absorption. The one in which we are interested today is absorption from the air.

It is important to remember the limitations inherent in the methods employed. Dr. Heyroth pointed out some of these as they relate to the transfer of results on animals, to human exposures. MAC is the sacred cow of toxicologic research, and has been accepted on inadequate bases for so long that one wonders if we should not use a different approach. We too often assume these allowable threshold values to be absolute in character. Nothing could be further from the truth.

The interpretation of the MAC is subject to a good many variables—the relevance of the data on which it is based, the compound employed, the kind of animal, and many other factors which Dr. Brandt and the author mentioned, or have suggested. Values are affected by the analytical method employed, since different analytical methods may give differing results. We know this to be especially true in the case of lead and arsenic. Application is further subject to the character of the data upon which it is based—upon the state of materials in the air, upon climate, perhaps, and upon the attributes of the population which is exposed. Agents' effect on men may be different from that on women; the effect on young people may be different from that on older people. My own opinion, in view of the many variables involved, is that if we are to think of a maximum allowable transfer as a threshold which we will apply by law or code, then we can do so only after human experience with a sufficient number of people to get a representative sample.

Dr. Heyroth has emphasized some of the difficulties of field observations. They cost time and money. It is hard to get adequate medical help. It is difficult to interest most people in a research point of view. I think, nevertheless, that we have to face

those difficulties if we are ever to evaluate the effects of the variables of absorption, distribution, partition, excretion, and response, with respect to the various attributes of industrial population.

I think Dr. Brandt is correct in his comment that we do the best we can with the data we have at hand. We should use it for what it is worth. I think in so using it, we ought not let it be translated in the form of a rigid code at the outset, but that rather we should leave the door open and initiate adequate clinical studies to develop a knowledge of effects upon humans in which we can have confidence.

W. N. WITHERIDGE, Detroit, Mich. (WRITTEN): Dr. Heyroth's paper is the best written commentary on the subject that I have seen. It should be read carefully by industrial hygienists, ventilating engineers, safety engineers, industrial physicians, general medical practitioners, toxicologists, chemists, physiologists, and numerous other workers in industry, insurance, government, law, education, and professional consultation.

This paper gives me an opportunity to present a matter of definition and clarification that has disturbed me for some time. Most industrial hygienists are aware of the fact that some of the discrepancies in the *maximum allowable concentration* values used around the country are due to a misunderstanding of the function of MAC. Perhaps the engineer who designs industrial-process ventilating equipment has been in a specially uncomfortable position in the midst of the confusion.

Unless a regulatory body or investigating agency is careful to specify, by the use of a list of MAC's, the objective it desires or the objective it has the authority to demand, there will continue to be conflicts across state borders, and between members of cooperating professions.

There are, in my opinion, the following possible uses and interpretations of MAC's of air contaminants:

(1) *The average concentration for day-long exposure of industrial workers that may not be exceeded without danger of physiological or pathological harm.* Most industrial hygiene agencies have this interpretation in mind when they establish or recommend MAC's for their jurisdiction. Frequently, however, the deliberations on the subject overlook the fact that a MAC for health maintenance or accident prevention may bear little relation to the MAC for uninterrupted production. Management and the industrial ventilating engineer want to know the latter values, but legislatures and common councils have not yet delegated to governmental agencies in times of peace the responsibility for maintaining efficient, economical, uninterrupted production. This is in the realm of management's competition for the labor and product markets. The right and duty of governmental investigating agencies to assure the community that industrial atmospheres are both safe and healthful is unquestioned by good management.

(2) *The average day-long concentration that can be maintained without causing widespread annoyance or excitement of the working force in the vicinity, resulting in turn in perpetual complaint, bickering, argument, threats of work stoppage, high labor turnover, wastefully long relief periods, and so forth.* Some will argue that such a concentration would be zero. Most of us are reasonable human beings, however, and a particularly offensive odor or irritant in our workplace that we might be able to overlook on occasion is nevertheless good cause for objection when it becomes a common occurrence. Good management is well aware that it cannot afford to suffer the economic loss caused by repeated or continuous atmospheric nuisance, and the ventilating engineer is often called in to improve the atmosphere around a process in cases where management cannot be cited for violation of law or ordinance.

Some nuisances are serious to such an extent that the law can step in. Others are practically impossible to eliminate through legal action, and the services of ventilating engineers are engaged simply because management has the intelligence (plus the money) to do everything within reason to maintain a nearly contented working force. Although there is available exceedingly little information on the nuisance value of air-borne substances used in industry, the ventilating engineer has discovered in many cases that he must stay far below the MAC for health, in order to preserve his good reputation. This item is important to A.S.H.V.E. membership, because the reputable and informed engineer will be careful in some cases to refuse to participate in any venture where the degree of ventilation for which money is available is only that which will satisfy the sovereign laws of the state.

(3) *The average day-long concentration that can be maintained without producing gradual inebriation of the workers, increase in their physiological reaction time, and consequent increase in the probability of accidents caused by impaired mental or muscular functioning.* The results are considered temporary for some volatile substances that have this effect, and in those cases any attempt to show definite disability under provisions of workmen's compensation laws ordinarily would fail.

This concentration is important to management interested in protecting all employees against the unsafe acts of a few, as well as safeguarding the exposed worker against accidental injury to himself. In this matter the employee is not always cooperative, because some volatile substances produce a pleasant level of slight intoxication sincerely appreciated by certain members of the working force. Employees have even objected to job transfer, change of solvent, or excessive ventilation, because of the sudden termination of respiratory stimulation. A few intoxicants are successfully metabolized and eliminated from the body without serious effects, and therefore, apart from the temporary effects more or less desirable according to viewpoint, they cannot be charged with causing prolonged or compensable occupational disability. But, for the same reason that management does not want employees in the plant who are under the influence of ethyl alcohol, they likewise do not knowingly expose workers to conditions that would have the same physiological effect.

(4) *The momentary, five, or ten-minute maximum concentration that can be tolerated by industrial workers without danger to health, and which, if exceeded because of failure of approved mechanical controls, would require immediate cessation of the offending process.* The importance of knowledge of such a limit is that production engineers may discover that the risk of mechanical failure of certain equipment using highly dangerous substances would be too great to permit its introduction into their plant. This implies, of course, that a choice of process or material is available. When the product itself is hazardous, as in the atomic energy project, the production engineers must devise elaborate multiple safeguards against mechanical failures and accidents.

The quickly dangerous concentration of an air contaminant is also of concern when an air cleaning device is considered for the salvage or restoration of ventilation air, thus making it possible to return the air to the plant after removal of toxic impurities. In some cases it will be evident that recirculation through an air cleaner offers too great a risk.

(5) *The maximum concentration that could be maintained for a period of a few minutes without driving all workers from the vicinity as a result of intense eye irritation.* No informed agency would intentionally propose such a limit for industrial atmospheres, but they can do just that if overly preoccupied with the task of preventing demonstrable systemic disease. This critical air concentration must be known for more substances, because in certain cases the maximum allowable concentration for health maintenance is too close to the maximum allowable concentration for eye tolerance to be used in ventilation design. Engineers usually want as a safety factor anywhere from two to five below the concentration that will cause condemnation of their equipment.

(6) *The maximum concentration that can be maintained for a few minutes without driving all workers from the vicinity as a result of intense nasal or respiratory irritation.* Comments under item 5 also apply here. Maximum tolerance concentrations for the eyes and respiratory passages are not identical for most substances, and the designer or investigator of industrial atmosphere controls should know, if possible, which is the lower value.

Although some persons insist that any condition that is uncomfortable is also unhealthful, workmen's compensation for occupational diseases cannot be adjudicated with justice on the basis of widely divergent claims of discomfort, when all attempts to show impairment of health or disability have failed. A *comfort* limit, which good management will maintain if at all practicable, cannot at this point in history be stated as the limit that governmental health agencies have authority to enforce. If the citizens of a state or city decide to grant such authority to their health or labor department, they must be careful to make provision that evidence to the effect that air concentrations in a given space are not below the comfort limit and does not automatically constitute legal proof that the atmosphere is also one that will produce compensable occupational disease.

(7) *The maximum concentration that can be tolerated without nauseating normal or healthy workers by a foul or sickening odor.* This is certainly a matter of some consequence to industrial physicians, for any working condition that precipitates nausea is one that should be promptly terminated. Although nausea in itself would be classified as a symptom rather than a disease, it may well warn of the possibility of exposure to a disease-producing agent. It is therefore in the realm of health maintenance to require elimination of any atmospheric condition that persistently or repeatedly causes nausea, even though the victims recover completely soon after each incident of intermittent exposure. Frequent gastro-intestinal upsets are also a strain on physical stamina, and industry cannot afford to subject workers to such depreciation. If management or investigators do not recognize their responsibility in this type of problem in air contamination, organized complaint or high labor turnover are the usual results.

(8) *The maximum concentration that is permissible at any moment as a measure of fire or explosion prevention.* Because of the conventional procedure of separating governmental inspections of factories between the functions of health maintenance and accident prevention, it is essential for the peace of mind of plant managers, plant engineers, ventilating engineers, and workers as well, that the upper limits of gas or vapor concentration allowed by safety organizations for fire prevention be clearly understood as being generally unsafe for day-long inhalation. For very brief worker exposures, the maximum for fire prevention and health maintenance might be nearly the same. These facts must be understood by all parties concerned so that the public can effectively insist upon close coordination of the functions of industrial accident prevention and industrial health maintenance.

Experienced workers in industrial health, occupational hygiene, air sanitation, and industrial toxicology know very well that the foregoing itemized possible functions of MAC's (selected according to one's viewpoint or occupational assignment) do not coincide at one figure for any of the known industrial air contaminants. The fact that we already have so much apparent agreement around the country on a large number of substances is more surprising than the fact that, also many discrepancies occur. A good deal of the agreement on MAC values, incidentally, is the frank result of the speed and ease with which we copy one another's data.

It is well known that many industrially-used substances have not been studied with all these criteria in mind. Most work has been done in physiological or toxicological laboratories to establish the relation between atmospheric concentrations and specific occupational diseases. It would certainly be helpful to the ventilating engineer if more studies were conducted on the sensory or perceptible responses to different concentrations of air contaminants, apart from any toxicological considerations.

My objective in presenting this discussion is to suggest that, whenever statements or tabulations of MAC's are made, written, or printed, the editor, author, or compiling group clearly indicate the *kind* of maximum that is being specified. Especially is this desirable when the values cited are not intended to cover all possible contingencies or objections. For example:

Maximum Allowable Concentration for Health
Maximum Allowable Concentration for Fire Safety
Maximum Allowable Concentration for Sobriety
Maximum Concentration for Eye Tolerance
Maximum Concentration for Olfactory Tolerance
Maximum Concentration for Respiratory Tolerance

The fact that many existing limits cannot be classified accurately according to such a scheme should be fair warning that they probably have no place in any compilation that is to be given the force of law. Those maximum limits which, after years of use and experience, are now known to give adequate but not luxurious or wasteful protection to health, life, and property, and at the same time to prevent the usual difficulties caused by highly disagreeable sensory responses, are the values that *can* be given a place in legal standards if the citizens of the district so insist.

May I express my sincere thanks to Dr. Heyroth for presenting to this Society a subject in which I am deeply interested, and which I believe deserves the thoughtful study of our entire membership.

L. E. SEELEY, Durham, N. H.: This is just a question concerning the toxicity test. I would like to ask Dr. Heyroth whether or not he has run tests with a combination of different gases. I ask the question because I have information which indicates that two different gases, both potentially lethal, and both below equal levels, might be combined to cause death. I wonder if that sort of thing has yet been investigated.

H. ESSIN, Cleveland, Ohio: I would like to ask Dr. Heyroth a question in regard to experimental procedure, if he wants to go into it; if not, perhaps I can ask him privately, afterward. He mentioned that the air flow rates in the animal cages might be quite high at times. I would appreciate his discussing briefly the criteria of selecting. I have in mind a high flow rate chilling an animal and bringing on the symptoms of cold; and too low a flow rate would not be adequate for respiration requirements. I would like to know how one goes about setting a figure for flow rate when running these experiments.

LESTER T. AVERY, Cleveland, Ohio: The question was asked, which I think is significant, by Mr. Edwards of Cincinnati, "Is this a meeting of the A.S.H.V.E. or the *American Medical Association?*"

Those of you who have heard me speak on this subject before will know that I think this is the most important subject the Society has to handle. The fact that we had over 1,000 men here to listen to the panel heating discussion, and that about half of that number are now on industrial hygiene ventilation, shows that you people do not think this subject is important to the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

As a background for that statement that it should be important to you, I remind you of the recent war in which our lives were at stake and were protected by the people who work in these environments—for instance, the people of Jack and Heintz, who were making the starters for the airplanes and the automatic pilots, the people at the Hoover Co., who were making the V-T fuse; and the people in any foundry who were making casings. You still have another interest as a taxpayer, because, as a taxpayer, you must pay for the waste of money which comes from the loss of time in industry, the hospitalization requirements, the old-age insurance which is necessary because a man cannot work. Sickness and accident rates are all established at your expense. Do not ever forget that this cost of industrial loss is yours, and you are paying for it.

This Society has in its membership, men who work in these environments. Some of us may be exposed to these hazards without knowing it. You may be employers of labor, and you may be subject to penalties. You may even be contractors, as are we, who have been asked, "How would you do this, and make this a fit environment?" You may be a manufacturer of equipment, who is asked to build or design something, even if it is of no interest to you, but some other company may make air purifiers or cleaners.

Certainly, the whole thing ties up in our attitude toward this subject. We have a committee called the Technical Advisory Committee on Air Conditioning in Industry, with W. L. Fleisher as chairman. Several years ago, in New York, he said, that *this subject is so big it is as big as the entire Society.* I am afraid the subject was too big for that committee. We have had very few meetings. Dr. Brandt has done an excellent job in holding a committee meeting last spring, and in October 1945 W. N. Witheridge held a meeting in Detroit. Incidentally, these men are assigned to different industries, Dr. Brandt to Bethlehem Steel, and Mr. Witheridge to General Motors, as experts to study their particular industrial air problems.

We have to depend on such fine medical men as Dr. Machle and Dr. Heyroth to present this kind of material. In our own group we do not have men who are competent to speak on the subject.

I suggest to you that, if this subject is too large for our committee, we break the committee down into several parts and get some parts small enough so that this Society can handle them. Bring in some constructive reports for our own membership so that this *will* be a meeting, Mr. Edwards, of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, with the full cooperation of the doctors with whom we have a common interest.

AUTHOR'S CLOSURE: There is, of course, at the present moment, a very considerable difference of opinion in regard to MAC. I think one has to remember that the entire field is undergoing growing pains. As I pointed out, it is only in the last 13 years that very much has been done along this line, and it is rather obvious that certain persons who might undertake such experimental work will become overenthusiastic and will attempt before they can walk, to run in setting codes.

I am thoroughly in accord with the idea that field observations and human experimentation are necessary. I did not have quite the time to devote to those phases as I did to the experimental work on animals, but people are of equal, if not greater importance.

Now, as to one or two of the questions. I was asked about combinations of gases. It is widely recognized that combinations of gases may behave quite differently. One might have a gas which is capable of stimulating the nervous system, leading to convulsions. One might also have a gas which is an anesthetic. If those two gases were mixed, we might not get anything, or we might get one or the other effect in varying proportions, according to the proportionate mixture of the gases and their relative activities. In the field of pharmacology a great deal of work is done on what is known as the synergism of the drugs, that is, the way in which two of them, having somewhat similar activities may together have a greater activity than the combination would imply.

There are also antagonisms but those matters are rather complicated and, unless we have a particular industrial problem in which we know we have an exposure that is going to involve two materials simultaneously, it is preferable to work out each of them thoroughly first and get that information before we begin to complicate the picture.

I was asked about how flow rates are selected. Well, it is obvious that on the one hand, one must have the rates sufficiently high to prevent death from asphyxia, and that, on the other hand we must assure ourselves that too high a flow rate does not, as the questioner indicates, lead to harm by increasing the incidence of respiratory disease. Usually, we conduct an experiment in which air, without the toxicant, is run through a chamber containing a parallel group of animals. That is, the experiment is run in parallel with another in which a like group of animals is subjected to air containing the toxic agent flowing at the same rate.

1310

No. 1310

MINIMAL REPLENISHMENT AIR REQUIRED FOR LIVING SPACES

Under Conditions of Mechanical Cooling and in Conjunction with the Removal of Odors by Activated Carbon and Other Means†

By WILLIAM V. CONSOLAZIO,* WASHINGTON, D. C.
AND LOUIS J. PECORA,** BETHESDA, MD.

INTRODUCTION

IN ENCLOSED living spaces, air conditioning simultaneously controls temperature, moisture content, movement and quality of the air.[1] In regard to air quality, it is generally accepted that when humans are the only source of contamination, the minimal quantity of outdoor air needed is that required to remove objectionable body odors and tobacco smoke. Under certain conditions factors such as oxygen and carbon dioxide concentration, bacterial content, and dust pollution also need to be considered. Since weight, space and cost of installation and operation are critical factors aboard ships, replenishment air for berthing spaces must be minimal if air conditioning is to become practical.

The basic problem considered in this report, therefore, is the determination of the minimum quantity of outdoor replenishment air necessary to keep odors in ships' berthing compartments within acceptable limits. The odor level in any space depends upon a number of factors, including structural characteristics of the environment, dietary and hygienic habits of the occupants, amount of outdoor air supplied, concentrations of tobacco smoke, the space allowed per person, capacity and temperature of the wet coil cooling surface, and the temperature and relative humidity of compartment and weather air.

---

†The material in this article should be construed only as the personal opinions of the writers and not as representing the opinion of the Navy Department officially.

*Head, Biochemistry Branch, Office of Naval Research.

**The National Institute of Health.

[1]Exponent numerals refer to Bibliography.

Presented at the 53rd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January 1947.

In the experiments to be described, a moderately air-tight compartment was utilized, and considerable care was exercised to reproduce the extreme conditions ordinarily encountered in berthing spaces.

Experimental Design and Procedures

Three groups of experiments were undertaken, as shown in Table 1. The first group (experiments 1-4) were designed to standardize operations, (*a*) to

Table 1. Experimental Conditions in Regard to Duration of Experiments, Dry and Wet Bulb Temperatures, and Replenishment Air Quantities

Experiment No.	Duration (hours)	Weather Air		Compartment Air		Replenishment air cfm/man	Remarks
		DB F	WB F	DB F	WB F		
1	144	......	......	86	73	15	78 deg ET
2	408	77.3	69.4	85	71	5	78 deg ET
3	120	81.0	73.6	85	72	5	78 deg ET
4	192	80.9	74.4	85	73	1	78 deg ET
5	192	78.7	69.9	86	73	15	78 deg ET
6	152	75.8	68.2	85	72	10	78 deg ET
7	192	75.6	67.4	86	71	5	78 deg ET
8	124	73.6	69.3	81	75	5	Same Effective Temp. WB and DB changed
9	124	72.8	68.1	87	76	53	81 deg ET Mechanical ventilation
10	105	67.5	64.2	86	71	5	78 deg ET No smoking
11	105	76.9	73.0	86	71	5	78 deg ET Ozone—Liquid Deodorizer
12	75	57.8	53.1	86	70	1	78 deg ET
13	105	58.5	53.6	89	74	1	82 deg ET
14	105	58.7	55.2	85	72	1	78 deg ET
15	105	64.3	58.3	85	72	1	78 deg ET
16	80	50.3	45.5	86	71	1	Conservation of air (carbon) 2250 cfm
17	96	50.2	44.7	85	71	1	Conservation of air (carbon) 2250 cfm
18	72	44.2	40.7	86	71	1	Conservation of air (carbon) 1100 cfm
19	72	37.0	32.8	84	69	1	Conservation of air (carbon) 550 cfm
20	48	47.5	46.0	86	71	1	No carbon
21	72	36.3	33.2	85	70	1	No carbon
22	120	39.6	36.5	84	67	1	No carbon, no occupants

indoctrinate the judges in odor perception, (*b*) to develop chemical or physical methods of odor determination, (*c*) to systematize the sampling of air for dust, bacteria and gases and (*d*) to establish a routine for the subjects. The second group (experiments 5, 6, 7, 12, 14, 15) was carried out under conditions of air cooling usually at 78 deg E T (85 F dry bulb—72 F wet bulb) at approximately 15, 10, 5 and 1 cfm replenishment air per man when the outside air was relatively warm and moist. At 53 cfm per man (experiment 9), mechanical ventilation was employed without cooling and the average compartment effective temperature rose to 81 deg (87 F dry bulb—76 F wet bulb). The third group (experiments 8, 10, 11, 13, 16, 17, 18, 19, 20, 21, 22) was designed to study the mechanics of odor development and control, (*a*) to determine the efficacy of activated

carbon and other agents designed to remove substances giving rise to odors, (*b*) to compare odor levels at various effective temperatures and (*c*) to study the effects of smoking and nonsmoking on the quality of the chamber air. Experiments 16-22 were performed during cold weather, when the heating of the air supply precluded condensation on cooling coil surfaces.

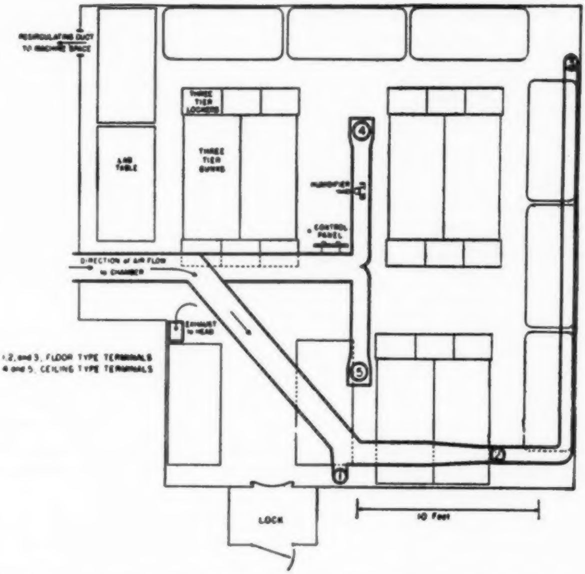


FIG. 1. LAYOUT OF EXPERIMENTAL COMPARTMENT. REPLENISHMENT AIR WAS DRAWN INTO THE RECIRCULATING DUCT AS IT ENTERED THE MACHINE SPACE BY WAY OF A *T* CUT INTO THIS DUCT JUST AFT OF THE COMPARTMENT PARTITION. THIS AIR THEN PROCEEDED OVER HEATING UNITS, THROUGH THE FAN, AND THEN OVER THE COOLING COILS WHEREUPON IT ENTERED THE COMPARTMENT WHERE THE EXCESS WAS EXHAUSTED INTO THE TOILET BELOW.

The Test Compartment: The dimensions of the test compartment were approximately 26 ft in length, 25 ft in width, and 10 ft in height (Fig. 1). The gross air volume was 6500 cu ft and the floor area approximately 650 sq ft. The floor, ceiling, and walls were constructed of sheet steel, with all joints lapped. An air lock was provided to minimize the effects of traffic to and from the chamber. The compartment, with its air conditioning machinery, was more airtight than a corresponding commercial installation. The space contained 45 bunks, lockers, and other accessories necessary for living purposes and accom-

modated 45 men who constantly occupied the compartment. The gross space per man was approximately 140 cu ft, which included the space occupied by furnishings.

Toilet facilities were in a separate compartment beneath the berthing space, and served the occupants of the experimental compartment. Its dimensions were approximately 13 ft in length, 11 ft in width, and 10 ft in height; the air space was approximately 1500 cu ft. It contained two toilets, one urinal, two wash stands, and one hopper. Shower facilities were available in an adjacent building.

Air Supply to Compartment: Mechanical supply and exhaust were provided, so that each occupant could be supplied with as much as 53 cfm. In addition, heating and cooling coils were available and provision was made for the recirculation of air and for the intake of measured quantities of outdoor air.

Two Navy Standard duct cooling coils (plate fin type) 12 x 30 in., with a face area of approximately 2.5 sq ft were used when air cooling was employed. Whenever there was use for the cooling coils, the surface temperatures were maintained between 40 and 55 F. Throughout the 22 experiments an attempt was made to keep the total air circulated over the coil surface as nearly constant as possible (approximately 2500 cfm). The ventilation practice of exhausting a berthing compartment through a duct into the toilet facility, and then outboard, was followed.

The replenishment air was introduced by means of a *T* cut into the recirculation duct; the intake consisted of a calibrated 12-in. diameter flow nozzle (coefficient = 0.99). All air flow measurements, including replenishment air, were made by a modification of King's linear hot wire anemometer.[3] The instrument employed was basically a wheatstone bridge, one arm of which consisted of a rigidly mounted section of 0.05 mm platinum wire approximately 1 in. long. This instrument was calibrated and supplied by the Material Laboratory, New York Naval Shipyard.

Replenishment air quantities were measured at the beginning, several times during the course, and at the end, of each experiment. Whenever the volume of replenishment air was changed, a complete air balance check was made of the overall air distribution which included recirculated air, replenishment air and exhaust air from the compartment.

Temperature, Humidity and Air Movement: A 78 deg E T (85 F dry bulb—72 F wet bulb) with an average air movement of approximately 30 fpm was maintained throughout most of the experiments. Exceptions were made in several experiments where odor levels were studied in relation to changes of temperature and humidity.

Water Vapor Content of Replenishment Air: The compartment temperatures were rigidly maintained by wet and dry bulb control devices of the conditioning machinery. In experiments conducted in the fall and winter of the year, the outside air had to be preheated before being brought into the compartment. During the cold periods, refrigeration was still supplied to the cooling coils to maintain the desired wet bulb conditions, and also as a control on earlier experiments.

Compartment Routine: Experiments were of from 2 to 17 days duration. Before a new experiment was begun, or whenever the replenishment air conditions were altered, all bed linens, mattresses and mattress covers were aired for at least 24 hours and the compartment, including ducts, coils, walls, and floors,

was thoroughly scrubbed. Discipline was maintained whenever it was essential to the experiment to keep the compartment fully occupied and the surroundings clean. The compartment was swept twice daily, and floors were washed once daily.

All soiled clothes, especially those used for work or athletics, were laundered only at a scheduled time. Subjects were allowed to change their underclothes as often as they desired, but they were required to stow soiled clothing in lockers within the compartment, in order to conform to conditions expected aboard naval ships.

Routine of Occupants: A total of 75 young men (18 to 22 years) served as subjects. Of these, 45 men were in the compartment at all times. Each man spent approximately 20 hours daily in the compartment, and four hours exercising outdoors. Exercise consisted of work or athletics sufficient in amount to produce sweating. A shower bath was required following exercise and before re-entering the compartment, which amounted to one shower per person per one and a half days. General toilet, such as washing the face and hands and brushing the teeth, was permitted early in the morning and after the evening meal. The men were free to visit the toilet facilities whenever they desired. Food was eaten at the regular mess in an adjacent building. With respect to diet, exercise, bathing and cleaning of clothes, the routine was similar to that aboard naval ships. Whenever occupants left the compartment to participate in group activities, stand-by personnel took their place. Smoking was permitted throughout all experiments, except on studies concerned with body odors.

A wide variation in habits of cleanliness and personal hygiene was noted among the subjects.

Estimate of Odor Levels: Although odor level was the critical determination in these experiments, there is no quantitative method available for recording it. It was necessary, therefore, to train men to judge degrees of odor on the bases outlined by Lehenberg et al[3] and Yaglou et al[4], modified to fit the conditions of the experiments. Sixteen trained men were employed to judge odors. The majority were smokers. In judging the quality of air, two odor impressions were made; the first (primary) was recorded immediately upon entrance into the compartment, the second (residual) was recorded after exposure of the judge to the environment for one minute. An attempt was made at identification of the odor or odors prevalent. The use of hair tonics, oranges, shoe polish, and other odoriferous substances interfering with odor perception was forbidden in the compartment.

The two odor impressions were obtained to satisfy the demands of the designing engineer. The question frequently asked is, For whom is the system being designed? If the system is being designed for the transient, then one set of conditions obtains; if it is being designed for the occupant, then another set obtains. In order to answer these questions, the primary impressions were obtained to gain the necessary information pertaining to the transient, whereas the residual impressions were obtained to gain similar information for the occupant. It is realized that the exposure of 1 min is not of sufficient duration to warrant such a conclusion. However, the error is in the right direction and the conditions favor the occupant.

The scale used to record odor impressions is shown on p. 132. Division is in half units, since experience showed that the trained judge could determine the

SCALE OF ODOR IMPRESSIONS

		Primary	Residual
NO ODOR	0.0	☐	☐
	0.5	☐	☐
THRESHOLD ODOR, JUST BARELY PERCEPTIBLE	1.0	☐	☐
	1.5	☐	☐
DEFINITE ODOR, BUT NOT OBJECTIONABLE (If I had the choice of theaters where all conditions other than this odor were the same, the odor would not prevent me from patronizing it.)	2.0	☐	☐
	2.5	☐	☐
STRONG OBJECTIONABLE ODOR (This odor is strong and sufficiently objectionable to prevent me from attending the theater in which it is present unless there were some sufficiently compensatory feature, *e.g.*, an excellent play.)	3.0	☐	☐
	3.5	☐	☐
AN ODOR VERY STRONG AND DISAGREEABLE, YET TOLERABLE (I would object strenuously to this odor in any place of occupation, yet it would not prevent me from working there if I knew it to be noninjurious.)	4.0	☐	☐
	4.5	☐	☐
AN OVERPOWERING, UNBEARABLE AND NAUSEATING ODOR	5.0	☐	☐

Instructions

1. Do not smoke for 20 min before entering the compartment.
2. Breathe deeply of fresh odor-free air for a few minutes before entering a test compartment, then enter quickly and with a few quick sniffs, decide upon and record the PRIMARY ODOR IMPRESSION. After one minute again take a few quick sniffs and record the RESIDUAL ODOR IMPRESSION.
3. Be certain that the Scale Value which you check as your PRIMARY ODOR IMPRESSION is the impression which you would receive if every sniff over the period of a working day were to be exactly as effective as the first one taken upon entering the compartment, i.e., avoid olfactory fatigue.
4. Keep thoroughly familiar with this Scale.

odor level to within half a unit. The judges were at no time conscious of the actual experimental manipulations, except in cases where the conditions were so obvious that the fact could not be hidden; for example, in the employment of carbon sorption units.

It is to be emphasized that, although a scale of odor impressions was employed, the basis for the differences recorded in the odor impressions were qualitative, not quantitative. Thus, a disagreeable but faint odor might be scored 3.0, whereas a strong but less offensive odor, such as tobacco smoke, might be scored 2.0. This concept of odor perception is in contrast to that of Yaglou, in which odor intensity is stated to be in accord with the Weber-Fechner law of physiological reactions, namely, sensation $= K$ log of stimulus. K varies with the type of odor encountered, when the odor is not complicated by other odors. Combinations of odors completely nullify the application of this law and since in spaces occupied by humans one usually encounters more than one type of odor, application of this law is questionable.

Consequently, the chief distinction made in these experiments was to determine whether or not the prevailing odor or odors within the compartments were objectionable or acceptable.

Chemical Method for the Determination of Odoriferous Substances: Since perception of odors is purely qualitative, and is further limited by the fact that any quantitative basis, if it exists, rests on a logarithmic relationship of stimulus (odoriferous substances present) to sensation, an attempt was made to develop a chemical† method of analysis using an acid-permanganate solution ($KMnO_4$) to oxidize organic matter in the air. The method employed was based on a similar method described by Lang et al.[8]

A given volume of air was drawn at a definite rate through a small heated reaction vessel (Fig. 2) containing a dilute solution of acid-permanganate. After cooling and making this solution up to its original volume, the percent transmission of the solution was determined and compared with the absorption

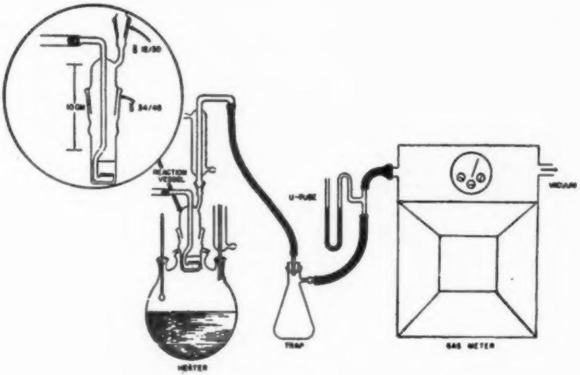


FIG. 2. APPARATUS USED FOR THE CHEMICAL ESTIMATION OF REDUCING SUBSTANCES IN AIR

of solutions of permanganate of known normality. The volume of air drawn through the scrubber was measured by means of a reinforced dry gas meter; an open U-tube mercury manometer was used to determine the pressure within the system. The flow was carefully regulated to allow 33 liters (1.165 cu ft) of air to pass through the scrubber in 20 min.

From a survey of the oxidation of various odoriferous compounds such as alcohols, aldehydes, ketones, ethers, esters, amines, mercaptans, and inorganic sulfides by acid-permanganate without drastically splitting the molecule, it was calculated that the number of atoms of oxygen required for oxidation of a molecule of the compounds averaged a little less than two. This gives little idea of the weight of the analyzed material suspended in the air, since the molecular weights of the compounds being oxidized are unknown. However, by expressing the final results in terms of milliequivalents of reduction of $KMnO_4$ occur-

†A physical system based on absorption of organic vapors on carbon with subsequent weighing failed to materialize into a reliable method.

ring per 1000 cu ft of standard air, there is obtained an empirical value which may be used for day-to-day comparisons.

The acid-permanganate solution is not reduced by a large group of strong smelling substances; namely the aliphatic acids, the hydrocarbons, and phenolic substances; however, tobacco smoke gives marked and rapid reduction of the solution.

Bacterial Assay of Air: The medium employed throughout the experiment was a meat infusion agar of the composition suggested by Schneiter et al.[6] All bacterial air sampling was done twice daily (10:30 a.m. and 1:30 p.m.) and at the same location.

Three methods of air sampling were employed.

1. *Petri Plate Method:* Sterile meat-infusion agar Petri plates were exposed for 15 min at levels of 15, 62, and 120 in. above the floor and equally spaced about the compartment. The resultant counts were averaged for the daily count.

2. *Impingement Method:* The adjustable funnel device as described by DuBuy et al, was used. The impingement rate was 25-27 liters (0.88-0.95 cu ft) per minute for a three minute period.

3. *Filtration Method:* The bubbler device, as described by Schneiter, et al[6] and DuBuy[7] was used. The rate of flow was 5-6 liters (0.18—0.21 cu ft) per minute for a 30-min period. Pour plates in triplicate were made from the sample after adjustment of the sample to the original volume with sterile water. The average of the triplicate plates was used for the daily average. For the detection of hemolytic streptococci, a duplicate set of pour plates was also made from the sample by the addition of 5 percent defibrinated rabbit blood to the meat-infusion agar.

The measurement of the rates of flow and volume of air samples was made by means of a dry gas meter and flow meter similar to that in Fig. 2. All plates were counted over a Quebec colony counter after 48 hr incubation at 37 C (98.6 F).

Dust Measurements: The Bausch and Lomb dust counter[8] was used throughout all these experiments in the assay of dust. Sampling and assays were done according to directions submitted with the instrument. Examination and counting were done by means of the attached microscope, using the counting graticule in the eyepiece. The average particle size was also determined by the use of special lines inscribed on the graticule for that purpose.

The samples were taken in the four corners of the chamber at three levels similar to Petri plate collection of bacteria, and at approximately the same time of day. Thus, it was possible to obtain a series of samples representative of the chamber as a whole.

Gas Sampling and Analysis: Twice daily, when air samples were taken for bacterial study, duplicate gas samples were collected for analysis of carbon dioxide and carbon monoxide content. Carbon monoxide analyses were made in the earlier experiments to determine whether or not smoking should be limited.

Sampling and Analysis of Condensate from Cooling Coils: At intervals corresponding to the time at which compartment odor recordings were made, condensate water was collected from the cooling coils of the air conditioning system. The quantity of water was measured, and its odor level was determined along with an identification of the odors present. On two occasions a bacterial examination of the condensate was made.

Employment of Substances to Remove or Mask Odor: A series of experiments was carried out at 78 deg E T to evaluate the efficiency of (1) activated carbon, (2) a liquid deodorizer, and (3) ozone for controlling odors. These experiments were performed late in the fall when little condensate was available from the cooling coils. In some cases, the replenishment air had to be warmed and water vapor added to maintain the conditions required.

1. *Activated carbon:*[9] The carbon units employed contained a total of 80 canisters made up into four units. These units were laid out back to back in the center of the compartment. The rate of air flow over the carbon bed and the total available carbon were varied under conditions in which smoking was permitted and limited. These experiments were all carried out at a measurable replenishment air of 1 cfm per man. The total reducing substances were measured in these experiments, and odor impressions were obtained for the same conditions.

2. *Liquid Deodorant:* A nationally advertised material, which is a suspension containing mainly a mixture of paraformaldehyde and essential oils, and which, it is claimed, purifies the air by means of chlorophyll, was used (see Bibliography[10, 11]). This compound was chosen, since it has the characteristics of many commercially available so-called liquid deodorizers, and also because it has been widely exploited for odor control in both civilian institutions and aboard naval vessels.

3. *Ozone:* The ozone used was manufactured by a commercial type of instrument for which, it is claimed, none of the obnoxious by-products such as oxides of nitrogen are produced.[12] The setting of ozone delivery was made according to the manufacturer's recommendations.

In one series of tests, the judges were exposed to the liquid deodorizer or to ozone in an adjacent room for a period of ten minutes before making an appraisal of the quality of the compartment air. In another series of tests, the judges recorded their odor impressions of the compartment wherein the test substances were being introduced by way of the replenishment air intake. In these tests, the odor level of the substances introduced was definitely noticeable at the air terminals in the compartment and just perceptible within the compartment itself.

EXPERIMENTAL DATA AND COMMENTS

Air Supply to the Compartment

Even though the greatest precautions were taken in making measurements of air flow and distribution, considerable difficulty was experienced. Terminal velocities of supply ducts were high, and enough encumbrances were in the path of the air stream to produce turbulence. Even though egg-crate louver straighteners were inserted in the return ducts, it was difficult to duplicate the measurements more accurately, as shown in Table 2.

TABLE 2. AIR SUPPLY AND DISTRIBUTION TO THE COMPARTMENT

Replenishment air desired (cfm/man)	53	15	10	5	1
Supply air through compartment inlets (cfm)	2362	2910	2358	2800	2319
Replenishment air (cfm)	2362	685	520	225	45
Recirculated air (by difference) (cfm)	....	2225	1838	2575	2274
Recirculated air (measured) (cfm)	....	1990	1730.	2683	1965
Error in measurements (cfm)	....	235	108	108	309
Percent error	....	8.1	4.6	3.9	12.5
Outgoing air at exhaust terminal (cfm)	....	472	297	180	53
Replenishment air (cfm/man)	52.4	15.2	11.5	5.0	1.0
Total air supply (cfm/man)	52.4	64.6	52.6	62.2	53.4

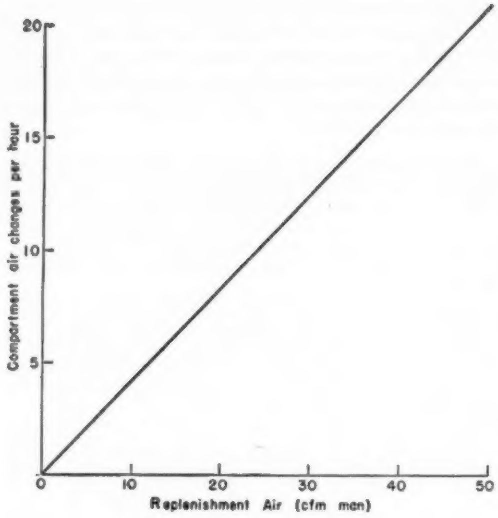


FIG. 3. RELATIONSHIP BETWEEN REPLENISHMENT AIR AND RATE OF AIR CHANGE IN THE COMPARTMENT

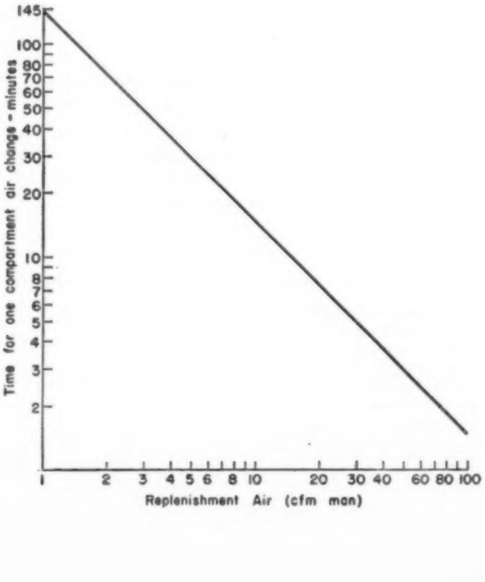


FIG. 4. RELATIONSHIP BETWEEN REPLENISHMENT AIR AND THE TIME FOR ONE COMPLETE AIR CHANGE IN THE COMPARTMENTS

From an engineering point of view, a ± 15 percent error is considered permissible in field work. These data show considerably less error than is usually acceptable in tests of commercial installations.

The graph showing the relationship of replenishment air to the compartment air, changes per hour, is presented in Fig. 3 and is linear, as it should be. Since the slope of this curve is related to the volume of the compartment, such a plot is essential in defining the experimental conditions. It is apparent that the dilution of any gas such as carbon dioxide or odors produced in the compartment at a constant rate should theoretically follow a straight line when plotted against

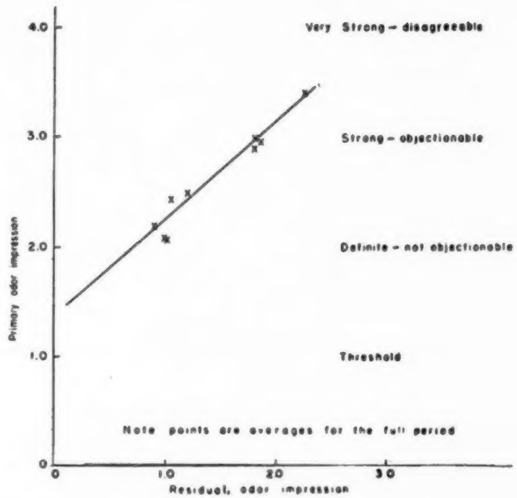


FIG. 5. RELATIONSHIP BETWEEN PRIMARY AND RESIDUAL ODOR IMPRESSIONS

replenishment air logarithmic coordinates. Consequently, in defining the compartment conditions the plot, replenishment-air against the time for one compartment air change, (Fig. 4), is much to be preferred.

Relation of Odor Level to Replenishment Air Supply

Relationship Between Primary and Residual Odor Impressions: There is good correlation between the two odor impressions (Fig. 5). That some fatigue of the olfactory receptors occurs during the first minute of exposure, is shown by the higher level of about 1.2 units recorded for the primary impressions.

Replenishment Air Supply in Relation to Primary Odor Level: At 78 deg E T (85 F dry bulb—72 F wet bulb) a base line could be drawn at an odor level of about 2.0 units (*definite—but not objectionable*) to include values from 10 to

53 cfm of replenishment air per man (Fig. 6). Actually, the odor level with mechanical ventilation, wherein as much as 53 cfm per man was introduced, was of the same order of magnitude as the odor level with air cooling at a replenishment air supply of only 5 cfm per man. The odors responsible for the level of the base line (approximately 2.0 units) emanated from bedding, mattress covers and other material in the compartment.

Upon reduction of replenishment air volume to measurable flows as low as 1 cfm per man, the score for the primary odor impression increased arithmetical-

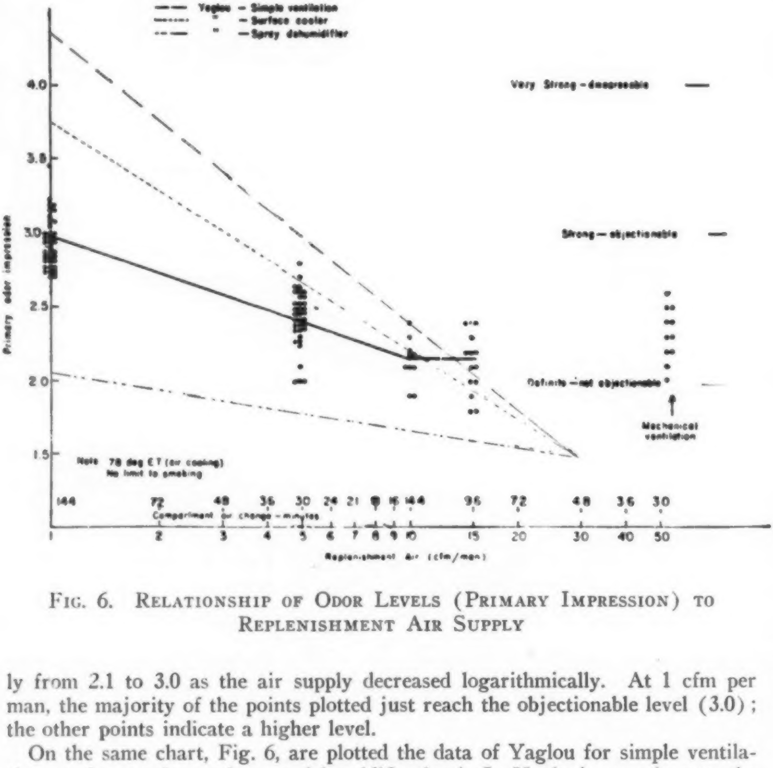


FIG. 6. RELATIONSHIP OF ODOR LEVELS (PRIMARY IMPRESSION) TO REPLENISHMENT AIR SUPPLY

ly from 2.1 to 3.0 as the air supply decreased logarithmically. At 1 cfm per man, the majority of the points plotted just reach the objectionable level (3.0); the other points indicate a higher level.

On the same chart, Fig. 6, are plotted the data of Yaglou for simple ventilation, surface cooler, and spray dehumidification.[4] In Yaglou's experiments, the available space per man was approximately double and the recirculated air approximately one-half that used in these experiments. From Yaglou's data (surface cooler) it is obvious that the compartment he employed must have been relatively free from furnishings, since the odor level at the higher replenishment air volumes approaches the threshold level (1.0) and his plot follows a straight line for approximately the same replenishment air volumes used in these experiments. In fact the compartment he employed was practically free from

furnishings.[13] Both plots show the same score for 10 cfm per man; otherwise the two plots are dissimilar. The odor levels recorded in the authors' experiments were considerably lower; in fact, our values for 1 cfm per man correspond fairly well with Yaglou's values for 3 cfm per man. It is to be borne in mind that the authors' experiments, for the most part, extended over a period of days, while those of Yaglou were only of several hours' duration, and that the experimental compartment was fully furnished and contained materials that were continuously emanating odors (fireproof mattress covers) as well as those that had high odor absorption characteristics, such as bedding. Another point of difference is that, in the authors' experiments, smoking was not limited and it may be

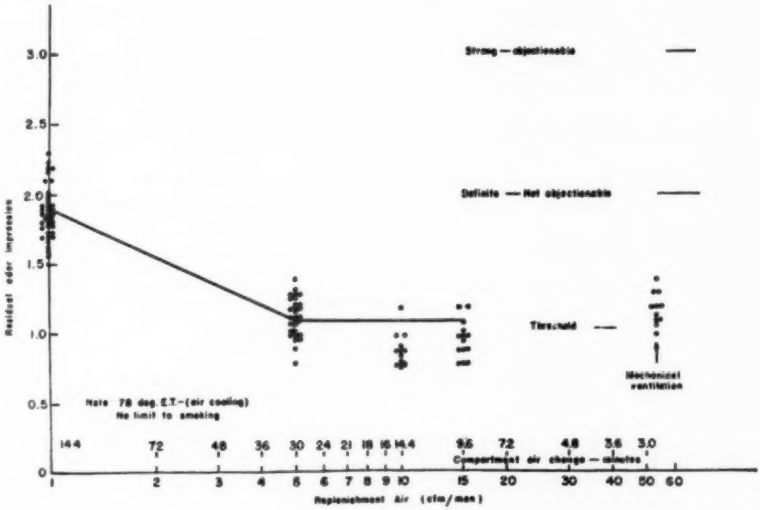


Fig. 7. Relationship of Odor Levels (Residual Impressions) to Replenishment Air Supply

that smoking had a masking effect on the type of odors that are more objectionable. Furthermore, since these experiments were performed during the summer months, considerable odoriferous matter may have been removed by condensation on the cooling coils and subsequently carried off along with the condensed water vapor. The compartment wet and dry bulb temperatures in Yaglou's experiments were always maintained at the comfort level and the weather conditions spread over the summer and winter seasons.[13]

Replenishment Air Supply in Relation to Residual Odor Level: For the same effective temperature, a similar type of curve is obtained as was the case in the primary odor levels. The residual odor impressions are at a much lower level (Fig. 7). When the air supply was reduced to 1 cfm per man (excluding leak-

age) the average odor impressions were just above 2.0 units, above the acceptable level. As already stated, these records were obtained in order to gain information as to the odor level that might be recorded by the individuals exposed to such an environment. It follows that, since exposure of the judges was of only one minute's duration, in comparison to the considerably greater exposure of the occupants, the scores attained would be of a higher order of magnitude than might be expected were the occupants to judge the quality of the air. In

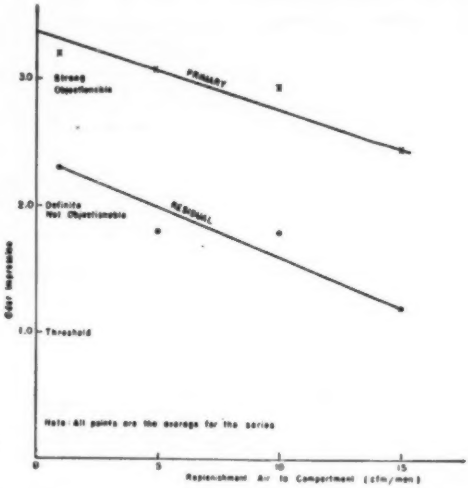


FIG. 8. RELATIONSHIP OF REPLENISHMENT AIR TO ODOR LEVEL IN TOILET FACILITY. ALL AIR TO THE TOILET WAS EXHAUST AIR FROM THE COMPARTMENT

fact, the occupants, when questioned, could find no fault with the compartment conditions—they could detect no odors.

Odor Intensity in the Toilet Facility: One must realize that this environment consisted of exhaust air from the compartment in equilibrium with the air in this space, and that the exhaust air from the compartment already had an odor level of its own which increased with the decrease in replenishment air volume. Consequently, the odor conditions obtained within the toilet facility were plotted against replenishment air to the compartment.

For the primary odor impression, the maximum value obtained was 3.2 units at 1 cfm per man (Fig. 8), slightly above the average odor level found in the main compartment for a corresponding replenishment air volume. For the residual odor impression the value was 2.3 units, which is above the acceptable level and only slightly higher than the level in the compartment. One point is emphasized, namely, that the primary odor level was objectionable at the lowest

compartment replenishment air volume of 1 cfm per man (approximately two complete air changes per hour) for the toilet facility, a level comparable to that recorded in the worst compartment condition. There is no reason to doubt that air flow to the toilet facility at the lower replenishment air volumes was non-existent, or that a reverse flow to the compartment may have occurred. Tests made at the entering and exhaust terminals to the toilet facility showed the necessary conditions were being obtained.

Effects of Temperature and Humidity on Odor Levels: For the same temperature of 78 deg ET, a decrease of dry bulb temperature from 85 F to 80 F does not appreciably alter the odor impressions. When the effective temperature, however, is elevated four degrees (78 to 82), there is some increase in the score of both primary and residual odor impressions.

Replenishment Air Supply in Relation to Reducing Substances in the Air: A series of experiments showed the permanganate method to be extremely sensitive for detecting the end products of tobacco smoke. On the other hand, it was only slightly sensitive in detecting agents that give rise to body odors. For example, at 5 cfm per man, the permanganate method gave a value of 4.1 meq of reducing substances per 1000 cu ft of air when smoking was allowed (primary odor level 2.5), and only 0.7 meq when there was no smoking (primary odor level 2.0). Such findings indicate that the permanganate method has its greatest value in estimating the degree of pollution caused by tobacco smoke.

Since the end products of tobacco smoke chiefly determine the results obtained by the permanganate method, and assuming that the amount of smoke production is constant, then the amount of reducing substances plotted against the volume of replenishment air should result in a straight line on logarithmic coordinates (Fig. 9). The reducing substance values, for the conditions between 5 to 15 cfm per man replenishment air, parallel the curve fairly well, leveling off below 5 cfm per man. However, there was very little difference in reducing substances between the 5 cfm per man (air cooling) and 53 cfm per man (mechanical ventilation) condition.

According to the log-log plot, the value for 1 cfm per man should have been several times the value found. These findings mean that the reducing substances were disappearing from the environment. Among the factors responsible may have been condensation on cooling coil surfaces and subsequent removal with condensed water vapor, condensation on walls and furnishings, or leakage within the duct work. The fact that the tobacco odor was very strong in the condensate water, together with the fact that coil surfaces and duct work were coated with a tarry-like substance with a strong odor of stale tobacco, indicates that absorption and condensation of the end products of smoking were in part responsible for the failure of the reducing substances to increase to the theoretical value. However, there is no doubt that leakage through duct work to adjacent spaces and through structural imperfections, and traffic to and from the chamber, played a considerable part at the lower air flows, even though the compartment employed was considerably tighter than the average installation. Yet if leakage were the main factor involved, then the primary and residual odor values should not have increased above the value for 5 cfm per man replenishment air.

It is of interest to note the marked increase in the amount of reducing substances found in the environment by the permanganate method of 1 cfm per man, when ducts and coils are not kept free from accumulated tar and lint. The

increase in value from 3.9 to 7.7 meq of reducing substances found is to be expected, since the more polluted the coil surfaces become, the more will air by-pass the coil surface and the less will be the amount of odoriferous substances condensed and subsequently removed with the condensate water. This increase in reducing substances emphasizes the need for coil and duct cleansers and the importance of the wet cooling coils as a means of odor control.

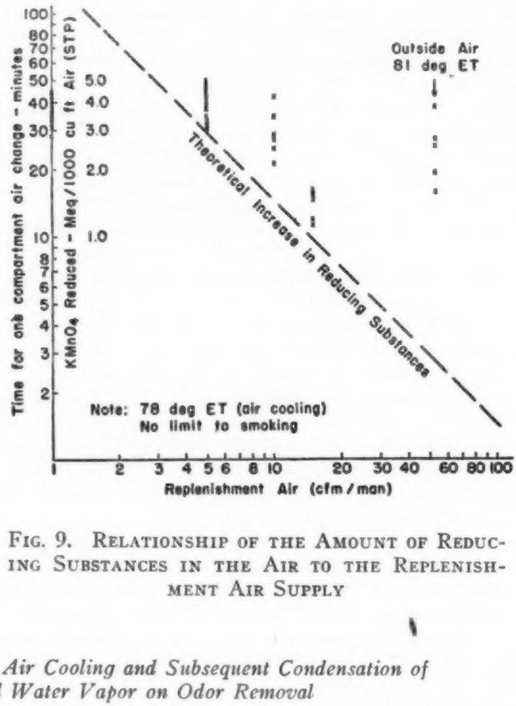


FIG. 9. RELATIONSHIP OF THE AMOUNT OF REDUCING SUBSTANCES IN THE AIR TO THE REPLENISHMENT AIR SUPPLY

Influence of Air Cooling and Subsequent Condensation of Odors and Water Vapor on Odor Removal

At 53 cfm per man and 81 deg ET, the characteristic compartment odors were those of stale tobacco, humans, metallic material, and bedding. As the replenishment air decreased to 5 cfm per man, the odor of stale tobacco became more and more prominent and there was a definite decrease in the odor of bedding and metallic furnishings. At the same time there was an increase in body odor. At 1 cfm per man, the odor of stale tobacco became less, whereas body odor became more discernible. It should be noted that the same types of odors were detected at 1 cfm per man with air cooling (78 deg ET) as at 53 cfm per man with mechanical ventilation (81 deg ET).

From the standpoint of odor types present in condensate water, the odor of stale tobacco showed a progressive increase with reduction in replenishment air,

whereas metallic or plumbing odors showed a decrease. Obviously, more and more odors due to stale tobacco were being removed as the replenishment air supply decreased and more air was being recirculated over the wet coils. Stale, musty, or body odor remained, practically at a constant level. These results substantiate the hypothesis that, in occupied spaces with unlimited tobacco smoking, considerable odoriferous substances are removed by condensation on the cooling coils and in conjunction with condensed water, and that the cooling coils can be used as adjuncts for odor removal.

The Influence of Activated Carbon in Reducing the Odor Levels

The system employed for cold weather operation was designed with cooling coil wet bulb control and dry bulb control of the heating elements. As already stated, the condensate normally obtained from the cooling coils in the earlier experiments was rarely obtained under these conditions. Humidification was employed to obtain the necessary wet bulb conditions. Consequently, it is not unreasonable to consider the following experimental conditions as equivalent to those obtained by mechanical ventilation. All experiments were carried out with 1 cfm per man replenishment air (Fig. 10).

Primary Odor Impressions: The same value, namely, 2.7, was found, whether the compartment was fully occupied or empty. The effectiveness of carbon in air recovery was demonstrated by the fact that the compartment was no more offensive when fully occupied under conditions which permitted unlimited smoking than the thoroughly scrubbed and aired, unoccupied compartment at 1 cfm per available bunk (experiments 16, 22). Odors detected during unoccupied compartment conditions were those of bedding, furnishings and the slight presence of stale tobacco. The effectiveness of carbon in reducing to 2.7 units an odor level that otherwise would have been 3.4 units, and objectionable, is shown by a comparison of experiments 16 and 20. It should be pointed out that as the available surface of carbon is reduced (experiment 19) or the amount of air treated decreases (experiment 18), a rise in odor level occurs. It is therefore probable that a better distribution of carbon units would make this method even more effective.

In addition, it was noted that the compartment air, which, prior to the use of carbon, contained a heavy hazy smoke, became clear and relatively free from haze when carbon was utilized. Furthermore, the complaints of eye irritation, under similar conditions of 1 cfm per man replenishment air, disappeared when carbon sorption was in effect. It may be that the felting on the carbon and its canisters caused by the high lint content within the compartment produced a filtering action on the tobacco smoke and aided in removing pyridine and other irritants. However, this possibility was not investigated.

Residual Odor Impressions: Under conditions of carbon sorption, values below 2.0 units were found for the residual odor impressions.

Reducing Substances as Determined by the Permanganate Method: There were fewer reducing substances in the air in those experiments where carbon sorption was used, and there was a progressive reduction in reducing substances when the carbon surface and air flow over the carbon bed were increased. From these data one may conclude that carbon, if properly employed, can become very effective in odor control under conditions of greatly restricted air supply.

*Condition and Maintenance of Activated Carbon Units:** After two weeks' use, all canisters and carbon were encrusted with a gray dust, which consisted chiefly of lint. Otherwise, both materials remained in excellent condition. The

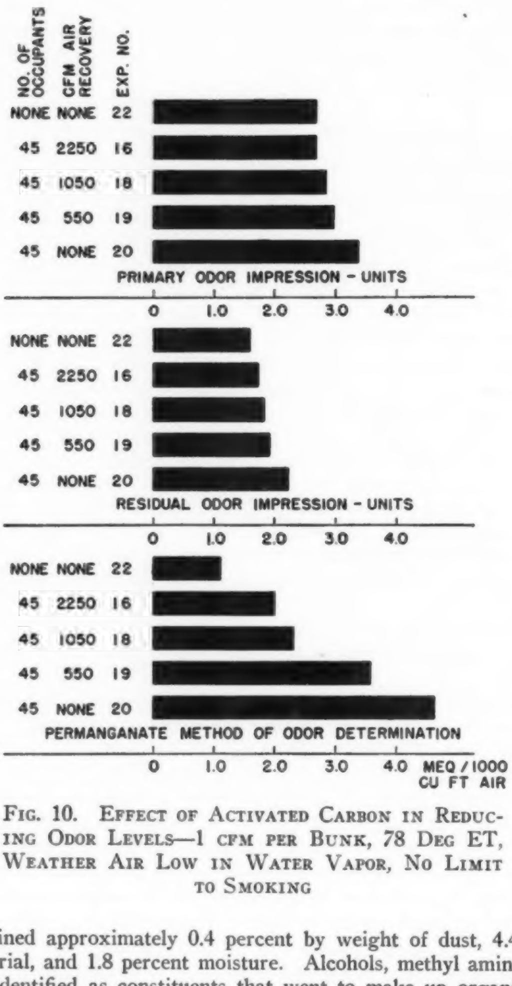


FIG. 10. EFFECT OF ACTIVATED CARBON IN REDUCING ODOR LEVELS—1 CFM PER BUNK, 78 DEG ET, WEATHER AIR LOW IN WATER VAPOR, NO LIMIT TO SMOKING

carbon contained approximately 0.4 percent by weight of dust, 4.4 percent of organic material, and 1.8 percent moisture. Alcohols, methyl amines and aldehydes were identified as constituents that went to make up organic materials. Minute quantities of sulfides, free sulfur or pyridines were also found on the carbon.

*Analysis was performed by the manufacturer.

Effect of Ozone in Reducing Odor Levels

Exposure of the judges to ozone for 10 min before entering the test compartment resulted in a decrease in primary odor score (Fig. 11). A possible explanation for this phenomenon, in agreement with the findings of previous investigators,[14, 15] is that a fatiguing or a paralysis of olfactory receptors had occurred. By contrast, in another series of experiments, when ozone was introduced into the test compartment without the knowledge of the judges, scores for both primary and residual odor impressions were slightly increased (Fig. 12).

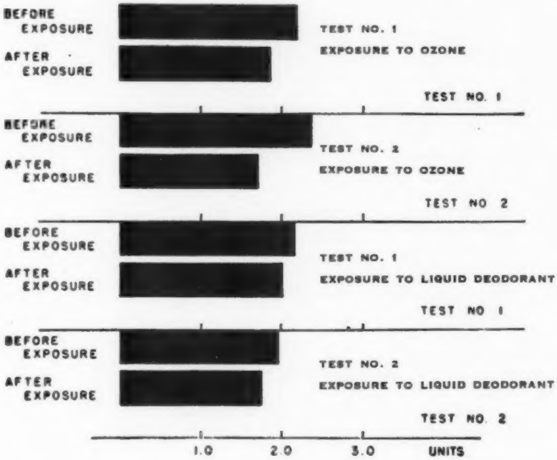


FIG. 11. INFLUENCE OF LIQUID DEODORANT AND OZONE ON ODOR LEVEL. 5 CFM PER OCCUPANT, 78 DEG ET. WEATHER AIR LOW IN WATER VAPOR

Even under the influence of masking produced by tobacco smoke, the compartment air was more disagreeable to the judges after ozone was introduced. It is probable that the reason for the findings of an increase in odor level may be explained on the basis that exposure time to ozone in these tests was too short for the fatiguing effect to be observed.

Liquid Deodorant

An exposure of 10 min to the liquid deodorant produced the same fatiguing effect on the olfactory receptors as was found for ozone (Fig. 11). The fatiguing effect, however, is not as marked as with ozone. It is of interest to record that when the liquid deodorant was used in the compartment, analysis of air by the permanganate method showed an increase from 0.7 to 3.7 meq per 1000 cu ft of air (Fig. 12). When, in addition, smoking was allowed under the same conditions, there was an increase from 4.1 to 5.5 meq.

The compartment primary odor level showed an increase when the liquid deodorant was introduced into the space, as it did for ozone. Consequently, it was concluded that, as in the case of ozone, preliminary exposure for approximately 10 min to the liquid type of deodorizer produces a paralyzing or fatiguing effect on the olfactory receptors.

The findings, namely, that *these substances do not control or oxidize odoriferous substances,* are in agreement with those of the joint publication of the *National Institute of Health* and the *National Bureau of Standards*[11] and corroborate previous screening tests of the authors.[10]

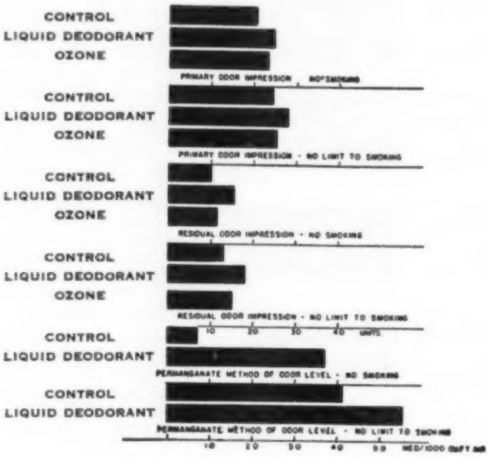


FIG. 12. INFLUENCE OF LIQUID DEODORANT AND OZONE ON ODOR LEVEL. 5 CFM PER OCCUPANT, 78 DEG ET, WEATHER AIR LOW IN WATER VAPOR

Changes in Bacterial Concentration in Relation to Replenishment Air Volume

It was found that the bacterial count remained at an extremely constant level under conditions of from 53 to 1 cfm per man of replenishment air at 78 deg ET in the compartment.

The settling velocity, as determined by the impingement and Petri plate counts, remained fairly constant throughout the experiments.

The incidence of hemolytic streptococci was practically zero in all experiments. This may have been due to the absence of respiratory infections among the subjects during the five-month period of air sampling.

Cultures on the condensate water from the cooling coils were practically sterile.

With the recovery of air by carbon (Table 3), there was a definite reduction in total bacteria count. This reduction may be due to the filtering action of the carbon and its canisters in removing lint and dust from the atmosphere. In experiment 21 a significant increase in bacterial count was obtained by removing the fireproof mattress covers and thus exposing the bedding. The increase in settling velocity and bacteria per particle indicates that this increase was due to dust and fabrics rather than to an increase in ventilation load by the occupants.

Concentration of Gases in Relation to Replenishment Air Volume

Carbon Monoxide: The carbon monoxide level due to tobacco smoking in confined spaces just reached 9 ppm (parts per million) at 1 cfm per man re-

TABLE 3. EFFECT OF MANIPULATION OF COMPARTMENT CONDITIONS ON BACTERIA COUNT AT A REPLENISHMENT AIR OF 1 CFM PER MAN

EXPERIMENT No.	TOTAL OCCUPANCY	COMPARTMENT TEMP		BACTERIA PER CU FT			REMARKS
		DB F	WB F	Plate	Impingement	Washing	
12	45	86	70	137	45	297	78 deg ET
13	45	89	74	180	75	287	82 deg ET
14	45	85	72	253	63	356	78 deg ET
15	45	85	72	220	61	263	78 deg ET
16	45	86	71	161	47	251	Conservation of air (carbon), 2250 cfm
17	45	85	71	167	42	174	Conservation of air (carbon), 2250 cfm
18	45	86	71	91	38	160	Conservation of air (carbon), 1100 cfm
19	45	84	69	87	50	198	Conservation of air (carbon), 550 cfm
20	45	86	71	94	42	234	78 deg ET
21	45	85	70	315	53	536	78 deg ET, no mattress covers
22	0	84	67	5	3	16	78 deg ET, no occupants

plenishment air, a level well below the tolerable concentration for everyday exposure. In fact, in some of our large metropolitan areas, the concentration frequently reaches 100-200 ppm.

Carbon Dioxide: The dilution curve of CO_2 produced at a constant rate should follow the log-log graph of Fig. 4. The analytical data, however, deviate from this curve, especially the values for 1 cfm per man (Fig. 13). Leakage is definitely a major factor in producing the deviation, although in a lesser degree, variation in activity of the occupants and absorption of carbon dioxide by condensate water and furnishings are also factors (see Bibliography[16]). It is at the highest concentration of carbon dioxide that all these factors would be expected to manifest themselves. The highest carbon dioxide concentration obtained (0.60 percent) was at the end of the first month of operations. At this time the subjects were most active, and activity had to be restrained. The lower points were obtained when the experiments were well underway, at which time most of the subjects were accustomed to the routine and consequently spent the greater portion of their time lying in their bunks.

The air cooling machinery was exceptionally tight, and all possible points of air escape were sealed with plastic putty. However, the recirculation of approximately 2500 cfm of air involved a high-pressure differential within the system, and consequently it was difficult to avoid some leakage. Opening and closing of doors, even with the use of an air lock, as in this case, constitutes a large leakage factor where more than 45 men are involved.

It is significant that the carbon dioxide curve shows the same characteristics as does the permanganate curve. Whatever factors influenced the permanganate

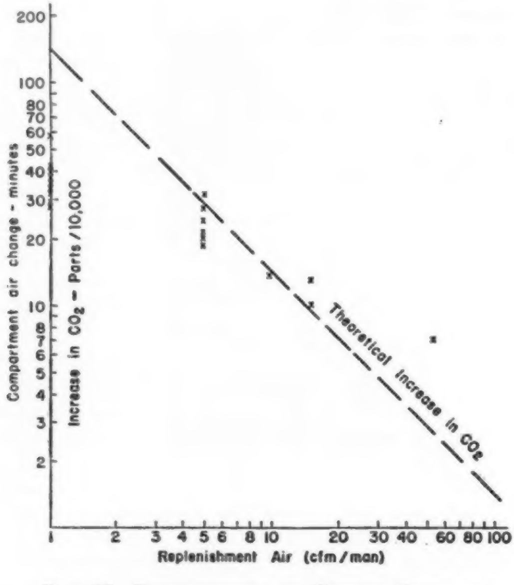


FIG. 13. RELATIONSHIP OF CARBON DIOXIDE CONCENTRATION TO REPLENISHMENT AIR SUPPLY

curve also probably were responsible for the characteristics of the CO_2 curve. However, neither of these curves showed any relation to the odor impression curves.

Concentration of Dust in Relation to Replenishment Air

The dust count remained fairly constant throughout the experiments, irrespective of replenishment air volume. The particle sizes ranged from 1-3μ (microns). Lint, of course, was a large factor in contributing to the content and has been treated elsewhere in this report.

Miscellaneous Comments

Source of Condensate Water: When the quantity of condensate from the cooling coils is plotted against the quantity of replenishment air (Fig. 14), an almost linear relationship is observed. Since, at 78 deg E T, some sweating is to be expected in active men, it is quite possible that the value of 3.5 liters (0.123 cu ft) per hour obtained (extrapolation to 0 cfm per man) may have been due primarily to loss in body water. If one assumes that 3.5 liters per hour is mainly from the occupants, then at 10 cfm per man, the increase of 6.5 liters (0.229 cu ft) may be due to the moisture from the weather air.

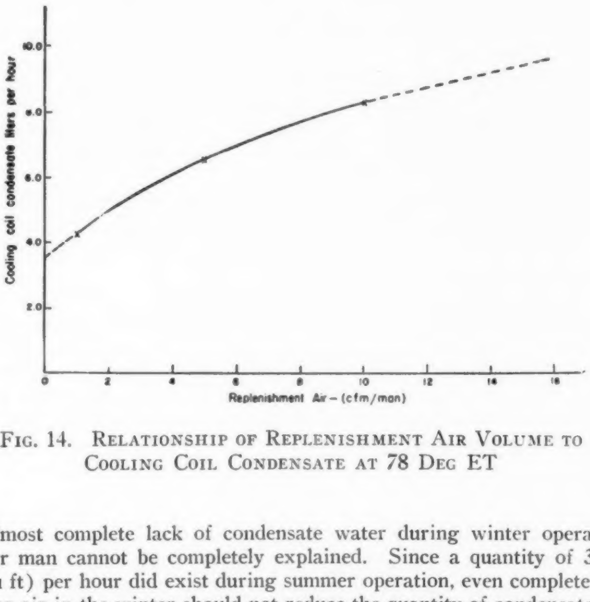


FIG. 14. RELATIONSHIP OF REPLENISHMENT AIR VOLUME TO COOLING COIL CONDENSATE AT 78 DEG ET

The almost complete lack of condensate water during winter operations at 1 cfm per man cannot be completely explained. Since a quantity of 3.5 liters (0.123 cu ft) per hour did exist during summer operation, even complete dryness of weather air in the winter should not reduce the quantity of condensate to such a low level. As previously stated, activity of the subjects was at a very low level during the winter months, and this factor might have been responsible in part for the lack of condensate water production. However, the continuous movement of the 45 occupants from a relatively high compartment water vapor content to a low weather water vapor content may have been one of the causes for the lack of condensate water. Air leakage through duct work also was a definite contributing factor.

Maintenance of Cooling Coil Surface and Ducts: After one week's operation of the cooling machinery at the lower levels of replenishment air volume (1 cfm per man), the coil and duct surfaces were coated with a foul smelling mixture of tar-like substances and lint, formed from substances of human origin and end products of tobacco smoke. The insertion of a 20-mesh screen filter into the

recirculation duct practically eliminated lint accumulation from the coil surface, but the accumulation of the lint on the screen necessitated its frequent removal and cleaning. Although the coil surface maintained most of its effectiveness when the screens were inserted, the coils and ducts continued to accumulate a thin layer of a tarry-like substance, which no doubt acted as a localized odor source.

With the introduction of a weekly spraying of the coil and duct surfaces with a dilute solution of tri-sodium phosphate, and subsequent rinsing with water, practically all traces of the foul smelling material were removed. This operation resulted in improved quality of compartment air. The fact that the permanganate level in the compartment at 1 cfm per man replenishment air was reduced from 7.7 to 3.9 meq per 1000 cu ft of air, when a tri-sodium phosphate spray and rinse was used, further demonstrates the necessity and effectiveness of periodic coil and duct cleaning.

Factors of Hygiene that Influence Cleanliness: In the earlier experiments, it was soon discovered that matters of hygiene and cleanliness could not be left to the discretion of the subjects. Accumulated debris, such as periodicals, dirt, cigarette butts and filthy clothes left in out-of-way places, made up a condition intolerable to fastidious individuals.

With the introduction of proper hygiene control and sanitation measures, this condition was soon remedied. It is quite possible that the constancy of dust and bacteria count and low level of odor encountered may have been in some way modified by such measures.

Vitiated Air: Frequently mentioned in the literature is the factor of *vitiated air,* wherein the occupants of such a space frequently undergo a loss of appetite and a disinclination to physical activity. The contrary was the case in these experiments. The subjects showed no loss of appetite, and restraint of the subjects was more of a problem than stimulation. As previously mentioned, at no time was there any complaint registered by the subjects indicating that the compartment environment was offensive.

SUMMARY

The essential problem in this investigation was to ascertain the minimal quantity of replenishment air required in a simulated ship's berthing space to keep odors at an acceptable level, in conjunction with air cooling. Furthermore, activated carbon, ozone, and a liquid deodorizer were evaluated as means of controlling odors.

The principal factors involved in these studies, with some comments, were the following: the quantity of replenishment air varied from 1 cfm to 53 cfm per man, gross air space was about 140 cu ft per man, and the recirculated air varied between 35 and 60 cfm per man; the temperature of the compartment air was usually 78 deg E T (85 F dry bulb—72 F wet bulb); in the summer months the amount of cooling coil condensate formed from the excess moisture in the outside air and the occupancy of 45 subjects varied from 1 to $2\frac{1}{2}$ gallons per hour, whereas no condensate formed in the winter months; the cooling coil temperature was usually maintained between 40 and 55 F in the warm weather, and a standard of personal hygiene and cleanliness of the compartment commensurate with shipboard routine was maintained.

The odor level for the condition of 5 cfm per man replenishment air at a temperature of 78 deg ET, with respect to the transient observer, was of the same order of magnitude as that for 53 cfm per man for weather air. Odoriferous substances were condensed on the cooling coils and subsequently removed, along with the water condensed from the air.

With activated carbon the replenishment air necessary for maintenance of an acceptable odor condition can be reduced considerably below 5 cfm per man. However, whether compartment leakage can be controlled so that dilution is less than 5 cfm per person is questionable, especially where large volumes of recirculated air are involved.

Ozone and the liquid chemical deodorant employed in separate tests did not reduce the odor level. Exposure to these substances, however, after a period of ten min apparently exerted an anesthetic effect relative to odor perception.

Carbon dioxide, carbon monoxide, bacteria, and dust did not reach objectionable or noxious levels, even with greatly reduced (1 cfm per man) replenishment air.

A chemical method, based upon the reduction of an acid-permanganate solution by organic matter in the air, proved to be of value in determining the degree of environmental pollution by the end products of tobacco smoke. The results, however, could not be correlated with the other odoriferous substances present, *e.g.*, odors of human origin.

Acknowledgements

The authors wish to express their sincere appreciation to Capt. E. G. Hakansson, (MC), USN, Capt. A. R. Behnke, (MC), USN, Prof. C. P. Yaglou, Messrs. Daniel Berres, George P. Lively, Welsh C. Whittlesey, T. H. Urdahl, Hugh T. Porter and E. Harry Culpepper for their timely advice and wholehearted support of the investigation; to Lt. Commdr. T. J. Duggan, H(S), USNR, and Chief Pharmacist's Mate J. D. Gillmore, USN, for their contributions to the bacteriologic phases of the investigation; to Pharmacist's Mates, USNR, N. L. Krueger, J. P. Hummel, R. T. Ottes, and J. A. Rae for their contributions of chemical and physical methods of organic vapor detection; to H. A. Collison, J. D. O'Neal, H. J. Mark, W. T. Platt and R. S. E. Smith for their contributions to the gas and dust phases of the problem; and to J. N. Shaner, W. R. Boehme, S. Hollander, and J. Rohrback for their contributions to the personnel phases. To those already mentioned and to Lt. Commdr. J. Hopkins, H(S), USNR, Lt. (jg) D. G. Doherty, H(S), USNR, and Ensign H. Schachman, H(S), USNR, the authors wish to extend their gratitude for the time and effort expended in judging the odor levels in the various spaces.

BIBLIOGRAPHY

1. Heating, Ventilating, Air Conditioning Guide, 1945, p. 787.

2. The Linear Hot Wire Anemometer and its Application in Technical Physics, by L. V. King. (*Journal Franklin Institute, 181*:1, 1916.)

3. A.S.H.V.E. Research Report No. 1009—A Laboratory Study of Minimum Ventilation Requirements: Ventilation Box Experiments, by W. H. Lehenberg, A. D. Brandt, and K. Morse. (A.S.H.V.E. Transactions, Vol. 41, 1935, p. 157.)

4. A.S.H.V.E. RESEARCH REPORT No. 1031—Ventilation Requirements, by C. P. Yaglou, E. C. Riley, and D. I. Coggins. (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 133.)

5. Determination of Spoilage in Protein Foodstuffs with Particular Reference to Fish, by O. W. Lang, L. Farber, C. Beck, and F. Yerman, *Ind. Eng. Chem., Anal. Ed., 16,* 490, 1944.)

6. Studies in Connection With the Selection of a Satisfactory Culture Medium for Bacterial Air Sampling, by R. Schneiter, J. E. Dunn, and B. A. Caminita. (*U. S. Public Health Report 60:* 789, 1945.)

7. A Comparative Study of Sampling Devices for Air-Borne Micro-Organisms, by H. G. DuBuy, A. Hollaender, and M. D. Lackey, (Supplement No. 184 to Public Health Reports, *U. S. Public Health Service,* 1945.)

8. Investigation of the Characteristics of the Bausch and Lomb Dust Counter, by S. W. Gurney, C. R. Williams and R. R. Meigs. (*Journal Industrial Hygiene and Toxicology, 20:* 24, 1938.)

9. Air Conservation Engineering, by G. S. Dauphinee, F. H. Munkelt, and H. Sleik. (W. B. Connor Engineering Corp., 1944.)

10. Evaluation of Six Commercial Means of Odor Control for Use in Inhabited Spaces, by D. G. Doherty and W. V. Consolazio. (Research Project X-533, Report No. 3, *Naval Medical Research Institute,* May 13, 1946.)

11. See report of *National Institute of Health and National Bureau of Standards.*

12. Influence of Nitrogen Oxides on the Toxicity of Ozone, by C. E. Thorp. (News Edition, ACS, *19:* 686, 1941.)

13. Personal communication from Prof. C. P. Yaglou. September 20, 1946.

14. Ozone in Ventilation—Its Possibilities and Limitations, by W. N. Witheridge and C. P. Yaglou. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 509.)

15. T. R. Crowder, Proceedings of the 20th Annual Meeting of Medicine and Surgery. (*Association of American Railroads,* New York, p. 123, June 11, 1940.)

16. Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw. (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 261.)

DISCUSSION

W. D. TURNER,* New York, N. Y. (WRITTEN): The authors are to be congratulated on their contribution of a fund of new data in the specialized field of ventilation dealing with odor control. This paper represents a scientific approach to a most difficult problem. The authors have produced new ideas in the matter of odor effects in living spaces, odor measurement, and odor removal.

Since we have been engaged for a number of years in intensive basic research in the field of odor counteraction, we were particularly interested in the comments in this paper concerning a liquid deodorant. Obviously, including this product in their comprehensive consideration, the authors state that, *these substances do not control or oxidize odoriferous substances.* It is not claimed that it controls or oxidizes odoriferous substances, whether malodorous or otherwise. Our explanation is that when the odor of this product is diffused into the air, it so opposes other odors present that the human nose experiences an effect of neutralization of both odors.

This principle of odor neutralization was established by the work of Zwaardemaker and others, and is mentioned in standard textbooks.† Examples of specific pairs of odors which neutralize each other in this way are bitter almond and musk, rubber and cedarwood, camphor and eau de cologne, and iodoform and Peruvian balsam. We have checked this neutralizing principle over a period of years on an extensive series

*Florida Chemical Research, Inc., New York, N. Y.

†Physiological Basis of Medical Practice, by Best & Taylor (2nd Edition, p. 1720). *The Chemical Senses,* by Moncrief.

of odors, and have determined that it is applicable with significant results in the case of ordinary occupancy odors. The application of this principle in reducing odor intensity is in definite contrast with the use of a strong odor to mask other odors, or the application of a principle of fatiguing or anesthetizing the olfactory senses.

The formulation of this liquid includes a low concentration of the complex extracted from the green plant cell, commercially produced and marketed as chlorophyll, together with a number of other components having characteristics which, when diffused into space, suggest a note of outdoor freshness. The effect of these substances is to give to air in living spaces an odor of low intensity which is more acceptable to the human breathing apparatus, and which is suggestive in some degree of the odor quality of air in the country.

That Dr. Consolazio recognizes the principle of odor counteraction as developed to commercial availability in the liquid tested, is evident from a statement that he made in a private conversation with the writer. He stated that he had advised the naval authorities for whom his research work was carried out that there were three ways to control the odor in ships' berthing compartments, namely:

1. By new air only, requiring 53 cfm per capita.
2. By partial recirculation through silica gel or carbon.
3. By use of odor counteractants such as the one he tested.

It is most significant that Dr. Consolazio's conception of human odor perception in occupied spaces is in contrast to the Weber-Fechner law of physiological reaction, namely, that sensation is proportional to the log of stimulus. As is pointed out in the paper under discussion, humanly occupied spaces usually involve more than one type of odor at a time. From this the authors conclude that the well established Weber-Fechner law is inoperative in determining human acceptability of odors in occupied spaces. This questioning of that law has such important implications that further research should be conducted to determine the validity of the law when so applied.

The paper describes a chemical method for the determination of odoriferous substances which is based on the effect of these substances on an acid permanganate solution. This test might be of some aid in establishing the day-by-day values of substances reducing the solution, provided no variables were introduced. This handicap is well recognized by the authors, throughout their paper. It would be difficult, however, to demonstrate any direct correlation between odor level and permanganate reduced, since permanganate either may be definitely affected by air of minimal odor value, or may be scarcely affected by air carrying a stench.

When this liquid is introduced into the air, a number of its constituents will themselves modify the acid permanganate solution. Hence, it might be erroneously inferred from the results of the permanganate tests that the introduction of the liquid resulted in the presence of increased amounts of malodorous substances. Therefore, this test cannot have validity in the appraisal of the value of liquid odor counteractants such as we have described.

The foregoing comments on this paper concern only those phases of the authors' work relative to the liquid deodorant as a liquid odor counteractant for living spaces. We repeat our earlier statement that the contribution of these authors is important with respect to a number of objectives within the study. It points up the difficulties that attend all effort connected with odor appraisals, and emphasizes the need for continuing research in this important aspect of control of air conditions in occupied spaces.

C. S. LEOPOLD, Philadelphia, Pa.: Did you keep a log of the number of people smoking at any one time?

A. R. BEHNKE, M.D.,** Washington, D. C.: No, that is a weakness in our test method. If we run these tests again we will do so. In other words, when the judges recorded their impressions, we did not know how many were smoking at the time, so absence of such data may give rise to the variation that you see between the scores that we got.

**Naval Medical Research Institute.

MR. LEOPOLD: I published a paper on the Relation of Fresh Air to Ventilation at Madison Square Garden. I made only a few checks of the number of people smoking, and I was amazed to find it was something less than 16 percent at any one time. I would say my total count was less than 500.

DR. BEHNKE: We did not count the number of cigarettes smoked, but when the replenishment of air was cut down to about one cubic foot per minute per man, there was enough smoke to cause eye irritation. Introduction of more outside air removed the irritation, removed the smoke and cleared up the eyes.

L. E. SEELEY, Durham, N. H.: Inasmuch as you have apparently no relationship between the treatment and the result of treatment on the individuals, I just wanted to know if the metabolism of the occupants did decrease at one cubic foot, because I think that might make a difference.

DR. BEHNKE: Yes, there was some decrease, because in the summertime, when the lower values were obtained, the individuals were less active. There was some decrease, but the order of decrease is not sufficient to account for what we saw. In other words, we should have obtained say 140 pp/10,000 of CO_2 and obtained between 30 and 60 parts which means that the activity would have had to be reduced to approximately one-third. It might have been reduced 20 percent but not to one-third of the initial level.

WILLARD MACHLE, M.D., New York, N. Y.: What is the correlation between the amount of reducing material and air?

DR. BEHNKE: That is the critical consideration. We do not know. If we use tobacco smoke, then there is a close correlation between odor level, as determined by olfactory perception and the amount of reducing substances found chemically.

Again, I want to bring out this point. It is hard to secure quantitative evaluation of odors. I doubt whether a correlation will ever be found between the amount of substances measured by a chemical method and the perception of smell, because what one non-smoking individual will call strong, at say a concentration of one to 1,000 of tobacco smoke, to a smoker will be acceptable, not strong. That is why we probably will not get a correlation.

W. E. ZIEBER, York, Pa.: I think this work is something in which we are all interested. Our organization carried on some experiments in this field, and we are very interested to know that this work included potassium permanganate methods of assaying the intensity of smoke, since we are attempting to do some correlation with this same chemical.

We are not in the same position as the Navy, which could confine a group of men to a compartment to obtain such fine data. So, our work is along the line of trying to burn a fixed amount of tobacco in a space within a certain time element and determining the effects upon the potassium permanganate solution. In that way, we calibrate a fixed quantity of solution for certain quantites of tobacco burned.

I think the authors might express themselves as to whether they think this method of evaluating tobacco smoke has any value. I will explain a little more in detail as to what method of attack we are using. We have weighed cigars and cigarettes in grams and burned a fixed number of grams of either one, as well as pipe tobacco. A quantity of air is passed through the permanganate solution. We expect to be able to determine, in this way, what reactions would be expected from the amount of tobacco present in the air to the equivalent grams burned in a fixed time element. One of our problems is that the visible part of tobacco smoke disappears rapidly and once this visible part disappears, the remainder is very rancid. These results are checked by filtering out the visible smoke. I do not know how much of this type of work Dr. Behnke has done, or how he has correlated this change of appearance of smoke with the various odors. I believe this is a very important factor. Dr. Behnke brought out the fact that the coils had to be cleaned once a week, which we have found very necessary in the past, especially if they were copper coils, not coated with any kind of protection. It would be interesting to know whether these coils were finned surfaces or plain copper, and whether they had any kind of tin or lead coating. Other questions are: Do the authors know the quantity of ozone placed in air? What method of measuring was used? I recognize that this is rather a major problem, and I am interested only from the standpoint of finding a method of determining this phase for our work.

Dr. Behnke: Answering Mr. Zieber, no measurement of the ozone was made. I do not know what the coil surfaces were like as I did not see them. The question of burning tobacco was discussed. It seems to be a good suggestion which we can try out in the Navy. We found that the products from tobacco after the smoke was removed settled on everything forming a thin coating on mattress covers, etc. and was responsible for the high initial odor level which could not be eliminated even by scrubbing the material.

It might be interesting to just capture that tobacco smoke residue and weigh it, and if such substances as dust and lint are excluded we might get a good idea of some type of correlation by such a rough method of collection.

L. S. Ries, Oberlin, Ohio: Is there a correlation between the moisture content of the air and the odor level? I noticed that no attempt was made at least it was not indicated of any dehumidification in this program.

G. A. Post, Indianapolis, Ind.: Was there a difference in the ability of smokers and non-smokers in differentiating between odors? Was there a connection between the taste?

Dr. Behnke: You are bringing out the weak points in this experimental method. I think that with some substances it made a difference, and some it did not. The idea was to get some judges who were smokers, others who were non-smokers. Some, of course, were sensitive to tobacco smoke; others were not. That gave a rather widespread opinion. I think it would be in the interest of accuracy to take a group of men who are not smokers, for example, and control it, that is, be more careful in the selection of the judges. The judges were purposely selected to include a wide variety of individuals, some who were sensitive to odors and those who were not.

Dr. B. M. Woods: I think we are getting around now to appreciate the significance of what Mr. Avery said a while ago. I was never more pleased than to have his comments on the significance of the subject of the morning.

The whole question of atmospheric environment, of course, is the one to which the Society has perhaps paid more significant attention over the years than any other, and it is the field in which the Society is basically interested. If the study of that question can be carried forward, and if we can continue the necessary collaboration with the medical men, together with the development of a group or groups within the Society competent to speak from the engineering point of view, I think we shall make a distinct advance. Certainly, it is to be hoped that we can.

Authors' Closure: Referring to Dr. Turner's comments, we are conscious of the publications relative to odor neutralization. However, we are not in agreement with the conclusions drawn from the point of view of commercial application. Our own opinions on odor control as forwarded to the cognizant naval authorities are that there are three mechanisms by which odor control may be instituted in berthing spaces; the factors to be considered in making the choice of methods are economics and attitude.

The mechanics of odor control are: (*a*) The maintenance of sufficient quantities of replenishment air, for the conditions as they now exist in the non-air conditioned berthing spaces, 50 to 60 cfm per man; for projected air conditioned spaces, 5 cfm per man. (*b*) By use of carbon sorption as now developed, or silica gel if and when it is demonstrated to be practical. (*c*) By use of odor masking agents. In this case the statement was qualified to take into consideration the esthetic and economic attitudes.

If it is decided that it is considered uneconomical to install means of removal of the odor, and that odor masking will suffice, then we see no need for the purchase of a proprietary item when the same masking may be accomplished by spraying the environment with any pleasant smelling essential oil. Furthermore, should we own a theater or assembly hall, and an odor problem confronted us, the simplest method of tackling such a problem, *if masking were my aim*, would be to allow the patron to smoke—in this way we would correct the odor problem from the patron's point of view, and at the same time create an atmosphere of sociability and welcome for the patrons.

Commenting on Mr. Leopold's inquiry, the average consumption of cigarettes was 20 per man per day, being highest at about noon and fairly constant until early evening, when it fell off. Odor impressions and analysis by permanganate method were lower in morning periods, and higher in the afternoon. No attempt was made to obtain a more specific survey. The introduction of air at 1 cfm per man while allowing the

occupants to smoke caused the complaint of eye irritation; the increase of replenishment air from 1 cfm to 5 cfm removed the cause of eye irritation and smoke, and cleared the air.

Replying to Dean Seeley, the metabolism decreased as the subjects became accustomed to the procedure and became somewhat bored during the progress of the tests. Metabolism had varied as much as 50 percent during certain periods of the day for early and late experiments.

Answering Dr. Machle's question, a correlation could be found between reducing substances in air and the amount of tobacco smoke, but not between odor levels if comprising odors from the body, furnishings, tobacco, etc. The lack of correlation is due to the finding that degree of odor, as measured by the individual, is not quantitative. While a slight quantity of butyric acid is offensive, a large quantity of vanillin or tobacco is acceptable. This fundamental concept must always be considered in analytical studies of odoriferous substances.

Referring to Mr. Zieber's comments, the coils were constructed of tinned copper tubes having tinned copper fins. A trisodium phosphate spray was used to wash the coil surfaces. The effect on condensation of odoriferous substances was not known.

The ozone quantity was not checked but was provided in accordance with manufacturer's instructions for operation of equipment used.

We believe that a definite correlation can be found between quantity of tobacco burned and reduction of permanganate. Since permanganate does not oxidize all organic compounds to CO_2 and H_2O, and since condensation of tobacco smoke and products occurs almost immediately, the factor of distance from the source and the time lag from consumption of tobacco to making of the analysis would certainly influence results obtained.

In regard to the question of Mr. Ries on effect of humidity on odor level, we found that while odor levels, recorded by the same judges, were higher in summer than in winter, we do not know whether the difference was due to a difference in humidity.

In reply to the question of Mr. Post concerning relative odor sensitivity of smokers and non-smokers, the subjects used were from a professional group and, as a group, were probably more sensitive than average. Fifty percent were about average, 25 percent were extremely sensitive, and 25 percent were of the tolerant type.

Until it is known whether design conditions are established for the tolerant individual, it is necessary to use the average although it may be advantageous to separate the test results of the more sensitive and the less sensitive groups.

1311

No. 1311

RESPONSE AND LAG IN THE CONTROL OF PANEL HEATING SYSTEMS†

By F. W. Hutchinson,* Lafayette, Ind.

Introduction

A FIVE ROOM residential-type structure, specially designed for use as a panel heating and cooling *laboratory*, has been equipped and is under test on the Housing Research Campus of Purdue University. The building, shown in Fig. 1, is not occupied and is reserved exclusively for use in carrying on basic research on the characteristics of panel heating and the relative performance of various types of panel systems.

Four separate panel systems are installed in each of the rooms. As shown in Fig. 2, a photograph taken during construction, sinuous coil type panels are located in the floor, the ceiling, the exterior walls, and the interior partitions. Each of these four systems can be used independently of the others and each has sufficient capacity to carry the entire heating load. Coils located in the floor are embedded in concrete whereas all others are embedded in plaster in accordance with usual construction procedures. Flow meters permit determination of the rate of circulation of hot water through any coil in any room while a totalizing meter gives the gallons per minute of warm water circulating through the system as a whole.

In the living room the ceiling coils are so arranged that alternate tubes discharge into different headers (see Fig. 3). By manipulation of valves in the lines discharging from the panel (valves shown below left center of Fig. 3) it is possible, in effect, to double the tube spacing in the ceiling. Further, a valve is located in the supply header to the living room ceiling in such a position (see Fig. 4) as to permit taking part of the panel area out of service. Thus the living room ceiling can be used as a heating surface of either of two

---

* Professor of Mechanical Engineering, Purdue University. Member A.S.H.V.E.

† A progress report resulting from a research project on panel heating being conducted at Purdue University under a grant from the *Copper and Brass Research Association;* This paper was prepared in cooperation with the Housing Research Foundation: G. Stanley Meikle, Research Director; Carl F. Boester, Housing Executive.

Presented at the 53rd Annual Meeting the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January 1947.

areas and of either of two coil spacings. For all tests discussed in the present paper the living room was used as the criterion of control and only the ceiling panel (full area with minimum tube spacing) was in service.

Energy for the heating system is provided from a gas-fired flash-type boiler as shown in Fig. 5. When the boiler operates in conjunction with a recirculating system using a three-way mixing valve to blend water from the boiler with water returning from the panels the flow rate through the boiler is, of course, a variable with load; under such conditions boiler water temperature is maintained at a fixed temperature (within the limits of the controller) and the burners operate on an off-on sequence as may be required to hold the boiler water temperature at the arbitrarily selected setting. Four solenoid-operated

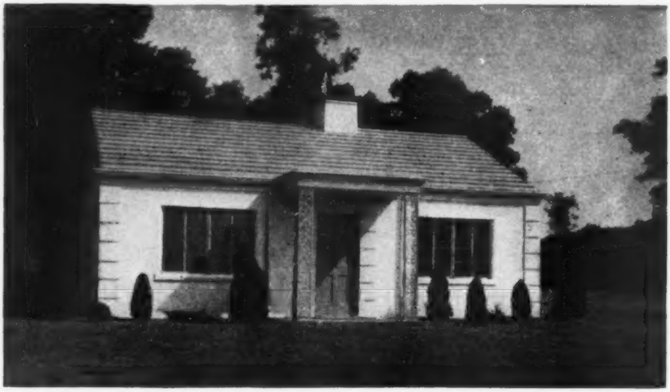


FIG. 1. TEST HOUSE FOR STUDY OF PANEL HEATING AND COOLING AT PURDUE UNIVERSITY

gas valves (two of which can be seen near the bottom of the boiler in Fig. 5) act simultaneously when the boiler is operating as described.

When the mixing valve is out of service and all water from the system passes through the boiler, control of the burners is under the direction of a thermostat located in the living room. For one type of control system, an off-on thermostat is used to simultaneously open or close all four gas valves whereas with another control system a modulating thermostat actuates the four valves in sequence. The cam shaft which controls the switches that operate the solenoid valves is shown on top of the boiler—just below the flue—in Fig. 5. In the same figure a flow meter is partially visible behind the flue and the indicating and recording meter for total flow of water to the system is shown on the wall. In all tests discussed in this paper the flow rates were 5 gpm and 1 gpm for the system and for the living room respectively.

All doors and windows of the living room were sealed except a door to an adjoining bedroom. Outside air was introduced into the living room at a rate of 1½ air changes per hour and escape of this air occurred through the unsealed door. The problem of control for the living room was accentuated due to the

fact that the front of the house faces south and two unshaded living room windows in this wall (see Fig. 1) receive unobstructed sunshine during all the morning hours and partially obstructed sunshine (due to grille supports of the portico) during the afternoon. The roof of the house is at an angle such that solar irradiation of the part over the living room—the south east section—is abnormally great. Hence the influence of solar load as a factor in determining panel heating control characteristics is probably greater for the living room of this test house than it would be for most rooms of an average house of non-solar construction.

Throughout the duration of all tests the temperatures of air, water, and surfaces were determined by means of thermocouples and recorded automatically. Fig. 6 shows the control station with a master panel for all thermocouples in the house and plugs to permit selecting the 32 thermocouples that are to be recorded—two of four available recorders are shown in the figure. Also evident in Fig. 6 is a manual potentiometer for checking purposes and to permit determination of readings from thermocouples that are not attached to the recorders; under the table is an ice bath which has been used in all tests in preference to cold-junction compensation. The control station is located in the *kitchen,* adjacent to the boiler room; when a test is in progress all equipment operates automatically so the operator need not enter any room of the house except to change flow charts once in 24 hrs; under no conditions does anyone enter the living room during the test period. Most tests have been of 60 hr duration and the shorter period selected for use in this report is in each case taken from a region near the middle of the 60 hr duration. Except where otherwise noted no test results are accepted until their consistency has been verified by at least one additional test period during which the same performance characteristic is observed.

FIG. 2. LOCATION OF PANEL SYSTEMS IN TEST HOUSE

Five Control Systems

The experimental work reported in this paper gives the performance, with living room air temperature at the breathing level as a criterion, of five different systems which were used for controlling the energy supply to the panels. The effectiveness of each system can be evaluated in terms of its ability to achieve a rapid rate of panel response and a minimum magnitude of deviation from the fixed control point. It must be emphasized that the purpose of this research has been to study response and lag factors in control of panel systems and that the selection of fixed air temperature as a criterion of effective control should not be construed as implying that fixed air temperature would be a desirable condition in a practical installation. These tests are *not* comfort tests

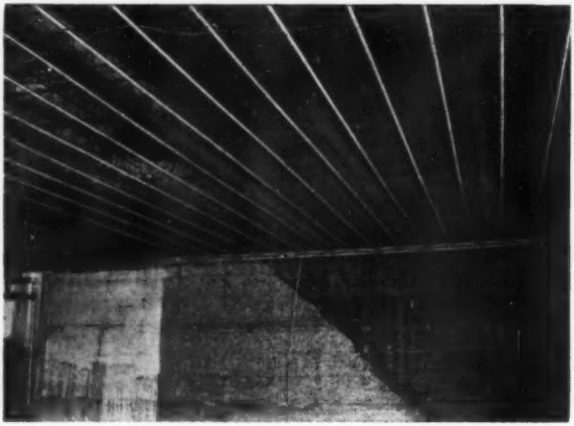


Fig. 3. Arrangement of living room ceiling coil for variation of tube spacing

hence it was not necessary to require that *comfort* conditions (necessitating variation of air temperature with load) be maintained within the room. For research purposes an adequate control system is one that will permit maintenance of arbitrarily assumed conditions; for practical purposes an adequate control system is one that will permit maintenance of the air temperature-mean radiant temperature relationship expressed by the Comfort Equation (sum of these two temperatures to be constant and equal numerically to 140 F). But any control system that will satisfy research requirements will also satisfy practical comfort requirements provided some device or method is employed to cause the control point (as expressed by inside air temperature) to vary as a function of load.

In some commercial equipments comfort conditions are approximated by resetting the inside air control point as a function of outside air temperature. This method assumes that outside air temperature varies as a lineal function of load hence is inexact due to neglect of the influence of wind effect, solar

irradiation, and internal load; the deviation is frequently not great and in some cases it appears to offer a curiously effective means of achieving what amounts, for certain seasons, to automatic night set-backs of the room air temperature.

Of the five systems tested three used living room air temperature as the basis of control, one used outside air temperature, and the other used a combination of inside and outside air temperatures. Thus,

I. Control Systems Actuated by Room Air Temperature

1. The first system—reported as Test No. 1—used an off-on thermostat actuating simultaneously all four gas valves ahead of the burner. With this arrangement,

FIG. 4. ARRANGEMENT OF LIVING ROOM CEILING COIL FOR VARIATION OF PANEL AREA

shown in Fig. 7*A*, all water going to the panels passes through the boiler; (the pump operates continuously for all systems).

2. The second system differs from the first only in that a modulating room thermostat replaces the off-on one and the four gas valves are actuated in sequence. Adjustment of cams permits spacing the control point of each gas valve at any desired point over the temperature range through which the thermostat provides modulation. The flow diagram for this arrangement is the same as for the first case (Fig. 7*A*).

3. Test No. 3 (Fig. 7*B*) is based on recirculation of water returning from the panels with parallel by-passing through the boiler of a fraction sufficient to provide a blend—at discharge from the three-way mixing valve—at whatever temperature may be needed to carry the load. Except for minor fluctuations due to difference in resistance between the main line and the by-pass, the total flow rate is constant, but the flow rate through the boiler varies with load. Temperature of water leaving the boiler is held at an arbitrarily fixed value by off-on operation of the burners under control of a temperature sensitive element in the pipe at boiler discharge. The three-way valve is actuated by a temperature sensitive element at discharge, the setting of which is reset, through relays, under the control of the modulating thermostat in the living room. All experimental work reported for Test No. 3 (as also for Tests

Nos. 4 and 5) is based on a fixed boiler water temperature setting at 120 F; the design value of this temperature for the heating system is 150 deg, but as the tests reported here were all run in periods of moderate load a more favorable three-way valve operating range was obtained by lowering the setting.

For the foregoing tests an arbitrary thermostat setting at 67 F was selected. With the thermostats used, however, it was found that the control point, for a fixed scale reading, varied somewhat between tests even though the average heating load was the same for the two test periods. To avoid confusing this

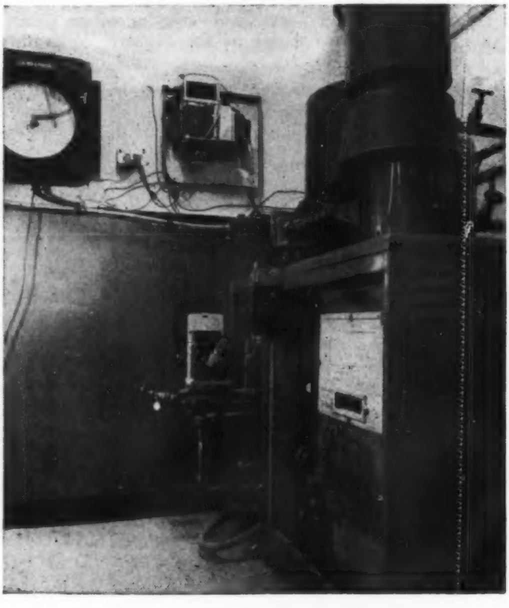


FIG. 5. BOILER AND CONTROLS

characteristic of the thermostat with performance characteristics of different control systems the experimental results have been slightly shifted in scale, for tests where this is necessary, to indicate 67 F as the room temperature setting. Since the doors of all rooms of the house were kept closed, circulation of air between rooms did not occur at a normal rate and maintenance of 67 F in the living room does not, therefore, mean that temperatures in all other rooms were at the same value. Actually, no particular care was exercised to adjust other rooms to a balanced condition, yet the temperature variations from room to room were not great. These temperatures have nothing to do, however, with the effectiveness of a given control system in maintaining a fixed air temperature within a particular zone hence they are not given further attention in this paper. In later tests consideration will be given to

the separate problem of determining the accuracy with which a five room one-story panel heated house can be treated as a single control zone.

II. Control System Actuated by Outside Air Temperature (Test No. 4).

This system is shown diagrammatically in Fig. *7B*, the outside temperature sensitive element replacing the room air thermostat in fixing the setting of the bulb which determines the position of the three-way valve. Experiment showed that on a cloudy day with 45 F air temperature the necessary temperature of water to the system was 105 F. The equivalent comfort temperature within the room was defined as

FIG. 6. TEMPERATURE INDICATING AND RECORDING STATION

70 F (allowing for a raised mean radiant temperature to offset the 3 F depression of room air temperature) so with 70 F outside air the water temperature should be 70 F. Assuming a straight line relationship between water temperature and outside air temperature (an assumption which implies that outside air temperature determines heating load), this would mean that at 0 F outside the required water temperature would be 140 F. Thus, a 70 F drop in outside temperature corresponds to a 70 F rise in water temperature and the controlling condition is therefore that the sum of outside air temperature and water (to the system) temperature must at all times be equal to 140 F. For the experimental work reported in Test No. 4 this desired relationship was not realized, so part at least of the departure of inside air temperature from the control point must be attributed to performance of the control system rather than to thermal lag in the structure.

III. Control System Actuated by Outside Air Temperature with Limited Re-Set from Inside Air Temperature (Test No. 5).

The flow diagram for this system is similar to Fig. *7B*, the outside temperature sensitive element setting the water temperature exactly as in Case II, but the inside air thermostat possessing re-set ability over a plus or minus 7 F range of water temperature. Thus a correction can be applied for solar load, wind load, or internal source load up to 14/70 or 20 percent of the total maximum heating load.

PERFORMANCE OF THE CONTROL SYSTEMS

Figs. 8 through 12, respectively, give test results from each of the five control systems which have been described. Representative test periods have been

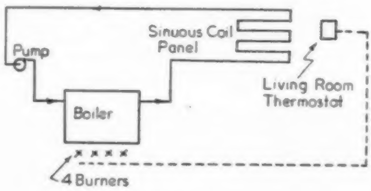


FIG. 7A

TEST NO. 1. OFF-ON THERMOSTAT ACTUATES FOUR BURNERS SIMULTANEOUSLY

TEST NO. 2. MODULATING THERMOSTAT ACTUATES FOUR BURNERS IN SEQUENCE

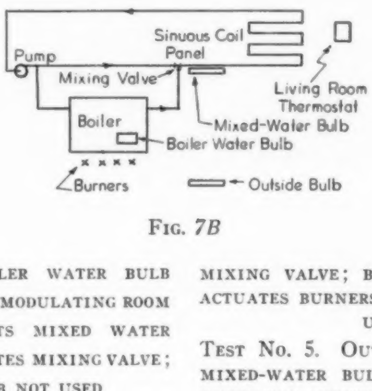


FIG. 7B

TEST NO. 3. BOILER WATER BULB ACTUATES BURNERS; MODULATING ROOM THERMOSTAT RE-SETS MIXED WATER BULB WHICH ACTUATES MIXING VALVE; OUTSIDE BULB NOT USED

TEST NO. 4. OUTSIDE BULB RE-SETS MIXED-WATER BULB WHICH ACTUATES MIXING VALVE; BOILER WATER BULB ACTUATES BURNERS; THERMOSTAT NOT USED

TEST NO. 5. OUTSIDE BULB RE-SETS MIXED-WATER BULB AND SETTING IS PARTIALLY CORRECTED BY ROOM THERMOSTAT; BOILER WATER BULB ACTUATES BURNERS

selected in each case and the same temperatures are plotted on each figure; these are,

1. Outside shaded air temperature, t_o.
2. Air temperature, t_a, 5 ft above the floor in the center of the living room.
3. Mean temperature, t_p, of the surface of the heating panel which occupied the entire living room ceiling. Thermocouples are installed on 2 ft centers over the surface of the panel and, although point-to-point variation does occur, experience with

the system has led to determination of a single thermocouple so located that its reading closely approximates the average of all the others.

4. Water temperature, t, of returning fluid.

5. Water temperature, t_g, of fluid going to the system. In every case this temperature was determined by a thermocouple located on the discharge side of the three-way valve; for tests in which all water passed through the boiler this reading would, of course, correspond to the boiler water temperature.

Test No. 1—Direct Water Heating with Burner Control by Off-On Thermostat: The data for this test (Figs. 8*A* and 8*B*) show remarkable constancy of room air temperature except during periods for which solar gain rapidly and materially reduces the load. Fig. 8*A* shows that during the twelve-hour period from 9 p.m. to 9 a.m. the outside temperature dropped 16 F, but the inside air temperature remained within 1 F (½ F for the greater part of the time) of the control point. Thus response of the system to sudden and substantial load change—as represented by the 8 F drop in outside temperature that accompanied a thundershower at 10 p.m. and occurred in less than 15 min—is rapid and does not allow undue departure from the control point.

From the standpoint of thermal capacity of the panel it is interesting to note that the lag effect is to provide an excellent damping characteristic. Thus, between midnight and 7a.m. the boiler cycles once in approximately 1½ hr with a periodic variation in leaving water temperature of 40 F, yet the corresponding periodic change in mean panel surface temperature is only ⅛ as great. The lag effect of the panel is clearly evident in that the maximum panel surface temperature usually occurs close to half an hour after the water temperature reaches its maximum value, that is, half an hour after the thermostat has been satisfied and has cut off the burner. If minimum panel temperature lagged minimum air temperature by a like amount the control would not be nearly so effective, but the figure shows that the heating-up curve for the panel (as for the water) is much steeper than the cooling curve hence response of the panel surface temperature to warmer water occurs instantly.

From the foregoing discussion it would appear that some optimum relationship exists between the available excess boiler capacity and the heat capacity of the panel. Since the excess energy output of the panel decreases as the heating load increases one would expect a flatter heating-up curve in cold weather and a greater lag in response of the panel surface to action by the room thermostat. Thus, in time of heavy load, the droop of panel surface temperature below the equilibrium value would be large and the overshoot of this temperature would be small, while in time of light load the opposite would occur.

This operating characteristic brings out an interesting fact with respect to the use of a fixed air temperature in a panel heated room. If perfect control were achieved neither panel droop nor overshoot would occur and by the conditions of comfort equation the room would be too hot under heavy load and too cool under light load. To establish comfort, with such a control system, the thermostatic setting should be lowered under high load and raised for low load. But, if this theoretically desirable change of setting is used with a cycling system of the type of Test No. 1, the room would be too cool during heavy load and too hot during light load. Thus, it follows that, fortuitously, the imperfection in control due to change, with load, in the slope of the heating curve serves to provide partial compensation, from the comfort standpoint, in a room

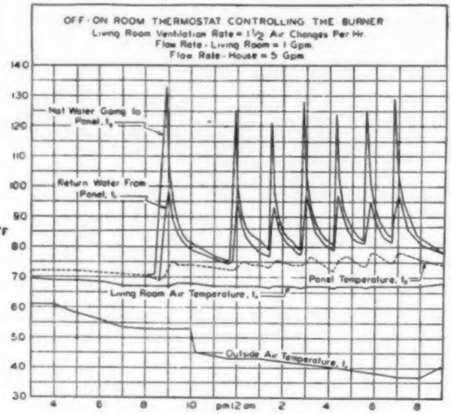


FIG. 8*A*. RESULTS FROM TEST NO. 1

for which the air temperature is not adjusted as a function of load. In any room with periodically varying panel surface temperature, conditions of optimum comfort cannot be realized, but the droop and overshoot characteristics will at least serve to lower the equivalent panel surface temperature under heavy load and to raise it during light load.

Note, Fig. 8*A*, that between 3 p.m. and 8 p.m. the heat was not on and the room temperature exceeded the control point. This condition probably is attributable to solar gain through the two unshaded living room windows.

Fig. 8*B* shows performance of the same control system when the outside air temperature remained practically constant over 24 hrs. Periodicity of panel

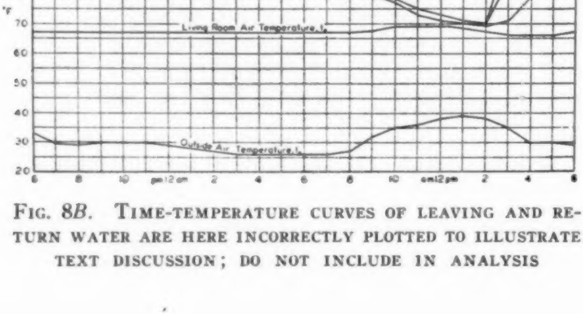


FIG. 8*B*. TIME-TEMPERATURE CURVES OF LEAVING AND RETURN WATER ARE HERE INCORRECTLY PLOTTED TO ILLUSTRATE TEXT DISCUSSION; DO NOT INCLUDE IN ANALYSIS

surface temperature no longer is as pronounced, but the influence of sunshine between the hours of 8 a.m. and 2 p.m. is very definite. The plottings of water temperatures in Fig. 8*B* have been made on an hourly basis and it is at once evident, by comparison with the water temperature curves of Fig. 8*A*, that an hourly plotting of periodic fluctuations is very unsatisfactory and may readily be responsible for erroneous concepts concerning performance of the system. Thus, in Fig. 8*B* the lines connecting hourly water temperatures imply alternate heating and cooling, within the temperature limits shown on the curve, of the boiler water. From a consideration of the shape of the curves of Fig. 8*A*, it is evident, however, that the 8*B* curves may be erroneous in magnitude, in slope, and in time of cycle. From an hourly temperature plot one cannot determine whether the temperature at each point is rising, falling, or at a point of inflection. Thus, when periodic variation occurs, the temperature-time plotting should be between points of maximum and minimum rather than for equal time increments; the water temperature curves of Fig. 8*B* are, therefore, shown here merely to demonstrate the hazards of such a plotting and they should not be used in analysis of system performance.

Test No. 2—Direct Water Heating with Burner Control by Modulating Thermostat: To reduce the amount of overshoot of panel surface temperature a modulating thermostat can operate the burner valves in sequence and thereby reduce the slope of the heating curve to any desired value. It will be noted, however, that excess slope reduction would result in droop of panel surface temperature with consequent over-cooling of the room. Fig. 9 shows that the slope of the heating curve with sequence operation of the burners is much less that when off-on control (Fig. 8*A*) is used and the slope always starts to fall off before the thermostat is satisfied; this is in contrast with the results of Test No. 1 for which the cut-off of the burners occurred during the period of high rate of water temperature rise.

The droop of panel surface temperature for the sequence control is not of great magnitude—averaging less than one degree—but the time lag is close to 15 min in contrast with the insignificant lag of Test No. 1. The degree of overshoot is approximately the same for Tests Nos. 1 and 2. The maximum variation in panel surface temperature for any one cycle is 6 F and since the panel area is roughtly ¼ of the total room surface, it follows that the probable order of magnitude of the corresponding change in mean radiant temperature is about 1½ F. The air temperature variation is about 1 F, but the minimums of air and panel surface occur simultaneously so the plus or minus departure of the equivalent comfort temperature (one-half the sum of air temperature and mean radiant temperature) is of the order of ¾ F. For the control system of Test No. 2 the heat was on for about 1/3 of the time in contrast to Test No. 1 (Fig. 8*A*) for which the on periods amount to 1/10 or less of the total time; the average outside air temperature for both of these tests was approximately the same, 50 F.

Of interest in the analysis of Fig. 9 would be knowledge of the time interval between sequence operation of the four burners. The setting of the cams which determine burner operation has already been described, but no on-off time record was kept for each burner and the heating curves do not permit assured estimate of the number of burners on at a given time, or the intervals between their coming on. For future tests of the sequence control system with floor

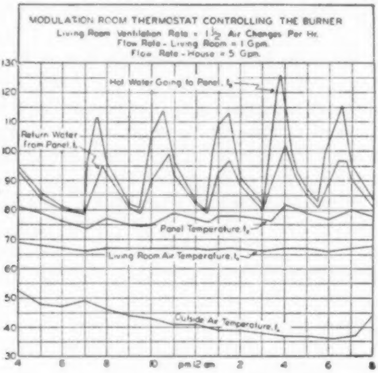


FIG. 9. RESULTS FROM TEST NO. 2.

and wall panels a procedure has been developed for obtaining a record of the off-on time for each of the four burners.

Test No. 3—Modulating Thermostatic Control of Water Temperature at Discharge from Three-Way Mixing Valves: The performance of the heating system under this method of control shows marked advantage, see Fig. 10, over the methods of Tests Nos. 1 and 2. With this system the overheating of the water and the periodicity of the panel surface temperature are no longer evident. Panel temperature now varies as a smooth inverse function of load as represented by the outside air temperature and striking uniformity of inside air temperature was observed throughout the test. The 22 hr test period shown in Fig. 10 covers an outside temperature variation of 30 F, yet no deviation

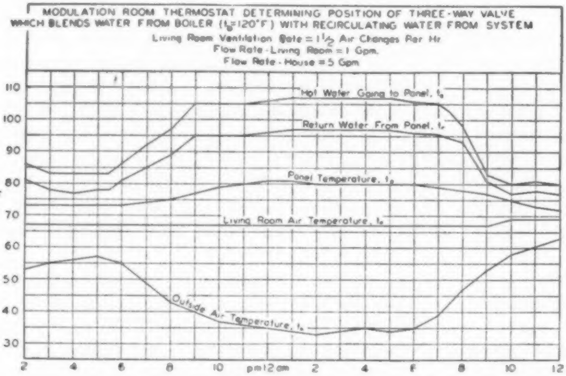


FIG. 10. RESULTS FROM TEST NO. 3

of inside air temperature from the control point occurred except during the last three hours of the test during which time the influence of solar radiation was sufficient to raise the room temperature even though the three-way valve was closed and no energy was being added to the house from the boiler. Note that in such circumstances overheating is not surprising since even though the heat supply is stopped the significant quantity of energy stored in the mass of the panel must continue to leave storage over a relatively long time period.

A greater number of tests of this type of control must be made before it will be possible to state whether or not the excellent performance shown in Fig. 10 is typical of what can be expected, but this one test and the few others which were made of the same system justify considerable optimism concerning the

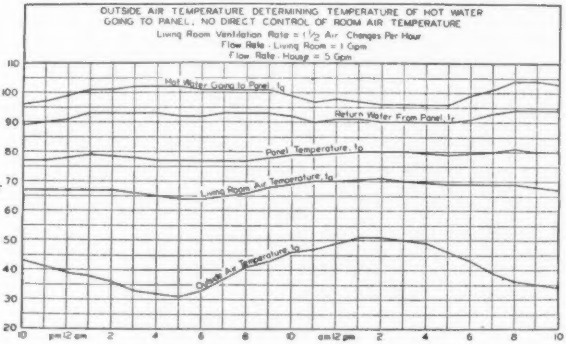


FIG. 11. RESULTS FROM TEST NO. 4

possibility of satisfactorily controlling a sinuous-coil plaster-embedded ceiling panel by means of a modulating thermostat and a three-way mixing valve.

Test No. 4—Outside Air Temperature Controlling Water Temperature at Discharge from Three-Way Mixing Valve: The results from this method of control, as shown in Fig. 11, are unsatisfactory since a ± 3½ deg departure from the control point occurred. As previously noted, however, part of this deviation must be attributed to the control equipment rather than to the control method since the desired linear relationship between outside air temperature and water temperature has not been realized. On the other hand the departure from this relationship which occurs during a long-time test is greater than that observable when the control equipment is separately checked for linearity hence there may be some influencing characteristic of the heating system that has not yet been recognized. By calculation from the data of Fig. 11 a correction to the observed inside air temperature can be made to take account of non-linearity of the control relationship; when such a calculation is made the resultant corrected air temperature curve is much more satisfactory than the actual one; this seems to indicate that the test results of Fig. 11 do not do

justice to this method of panel control. Further studies of this control system will be made and the results reported in a later paper.

Test No. 5—Outside Air Temperature Controlling Water Temperature with Partial Correction Available from Modulating Thermostat: Performance of this system is satisfactory (see Fig. 12) except during periods of sunshine simultaneously with moderate outside air temperatures; during such intervals the rise in room air temperature amounts to as much as 4 F. Part of the explanation of the solar rise probably is the reduced effectiveness of room air temperature change to vary the heat input required by changes in load other than that which is represented by the outside air temperature. Thus for the control system of Test No. 3 a sudden change in load occasioned by sunshine or by large change in occupancy can be reflected in a change of heat input over the range corresponding to zero to full design load. With the control system of Test No. 5, however, the room air temperature can exert a maximum influence

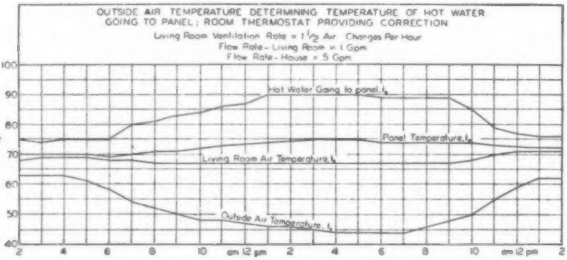


FIG. 12. RESULTS FROM TEST NO. 5

on the heat input of only a fraction of the total range hence the system: (1) May be unable to handle large auxiliary load changes; (2) Will require more time, hence permit greater deviation, in restoring equilibrium conditions once a disturbance has occurred.

On the favorable side, the control system used in this test provides an anticipatory feature which may be of considerable value in buildings where either the structure or the panel is of high thermal capacity and low thermal diffusivity. This application of the system will be investigated in connection with tests, now scheduled, using the concrete floor panels in the Purdue test house.

SUMMARY

This paper describes a panel heated test house which is in service on the Purdue University Research Housing Campus. The five-room residence is equipped with four independent sinuous-coil type hot-water panel systems located in the floor, ceiling, exterior walls, and interior partitions. Five different methods of controlling the panel heating are installed and have been tested. All tests reported in this paper are for ceiling panels *only* and were conducted with constancy of living room air temperature as a criterion of control; if

optimum comfort had been the criterion, the inside air temperature would have had to be varied as an inverse function of load.

Test results show periodicity of panel surface temperature for both of the control methods which use off-on operation (or modulating sequence operation) of the burners as a means of regulating the energy input to the heating system. The period varies with load and with control method, but is of the order of two hours per cycle with a change in temperature of the surface (at constant impressed load on the structure) of approximately 6 F. When control methods are used which permit regulation of energy input by blending recirculated water with boiler water at fixed temperature, the periodicity no longer occurs and stable control is, in general, realized; of three systems of this type which were tested, best results were obtained from the one using a modulating room thermostat to fix the setting of a temperature sensitive element in the discharge line from the mixing valve; this element then controlled the position of the valve.

Further control tests will be made during the present heating season using the same five control methods, but applying them to floor, wall, and partition panel heating systems. Additional tests on the ceiling system will be run to provide data from a greater range of load conditions; results will be given in a research report which is planned for release in May, 1947.

ACKNOWLEDGMENT

Thanks are due to Messrs. W. P. Chapman and J. M. Ayers for the transposition of test data and the preparation of final curves; both men are graduate students and research fellows at Purdue University.

DISCUSSION

F. W. CHAMBERS, Toronto, Ont., Canada: I would like to know about the results of the tests on outside wall panels. I am very interested in outside wall panels because I believe that the proper place to wage the battle between the defending heating system and the enemies, cold and discomfort, is in the outside walls, not inside our buildings.

PROFESSOR HUTCHINSON: The results, as far as our project on control was concerned, were practically the same. We achieved most effective control by means of a simple room thermostat, almost equally good control by means of outside control compensated by room thermostat, and relatively poor control by outside compensation alone.

A. A. ENGELBACH, Washington, D. C.: Did you have any instrumentation of the sun load at all? Also, what is the depth of the barrier in materials in the floor slabs?

PROFESSOR HUTCHINSON: We have no data which would provide any information on solar load. The outside controller was located in a shaded position. The house had southern exposure, no shades and no drapes. There were three windows in the south wall of the living room. Half of the sloped roof faced south, so that the sun's rays struck it at normal intensity during mid-day.

With respect to the second question, the coils were embedded in 2 in. of concrete under a ⅛ in. hardwood parquet floor.

MR. RICHARDS, Pittsburgh, Pa.: Is there any significance in the fact that the difference between room temperature and panel temperature increased, as shown in Fig. 12? Is this difference considered to have any physiological advantage?

PROFESSOR HUTCHINSON: The reason for the increase in Fig. 12 is because of greater load. In this case, the difference between room air temperature and panel temperature operating in the living room was 67 F. The panel reached a maximum of 75 F. That difference, providing we maintained fixed air temperature, would obviously be dependent on the outside air temperature. The panel temperature increases with load and is an index of the mean radiant temperature; therefore, the air temperature must decrease with load.

C. E. A. WINSLOW, New Haven, Conn.: I was struck with the same point the last speaker raised, and I should think that the condition represented in Fig. 12 was better than the one in Fig. 11. Now, of course, we do not know what the temperature of cold walls was. That varies too. In general, I would think this latter condition, in which the air temperature fell as the panel temperature rose, would be much more desirable than one in which the panel rose and the air temperature did not fall. This looks like an extremely neat and satisfactory condition.

PROFESSOR HUTCHINSON: One point that may lead to misunderstanding is that all tests were run at fixed air temperature setting; the rise in air temperature at both ends of the test was not a matter of control, but was due to solar load.

I fully agree with Dr. Winslow that as the panel temperature goes up a decreased air temperature is desirable. In this test report, the increase at both ends is not due to control, but to the fact that we lost control because of the high incidence of radiant energy.

W. A. DANIELSON, *Memphis, Tenn.*: I was wondering whether the concrete was laid directly on the ground, or not, because that would affect the heat storage in the ground. I was struck with the importance of this storage when it was 23 deg below in Minneapolis. The report was that the heat storage in the ground was drawn on to keep that temperature up.

I was wondering also if, when the heat panel was heated, the surface temperature of the floor rose. Manifestly the radiant energy must be absorbed some place and become sensible heat.

PROFESSOR HUTCHINSON: With respect to the second question by General Danielson, when we were operating with the ceiling panels the floor temperature was in all cases slightly above the air temperature in the room. That was in agreement with theory, and we did have some warming of the floor, irrespective of the location of panels. Whether the panels were exterior ones, or ceiling, there was warming of the floor in every case.

Since I was not there when the house was built and the coils put in, I will ask Mr. Boester if he can answer the first question.

C. F. BOESTER, Lafayette, Ind.: The house had its footings put down to a depth of 40 in. around the perimeter of the house. Then, directly on the grade without disturbing the soil, except the top soil, 8 in. vermiculite concrete was put down and 4 pipe coils were laid on it. This was covered with 2 in. of concrete and ⅛ in. wood parquet floor.

MR. DANIELSON: Did you put thermocouples in between ground and concrete and between concrete and the floor, and above the floor? That was the point raised previously.

PROFESSOR HUTCHINSON: No such data were obtained.

MR. DANIELSON: What was the theory on making up the poorer concrete, one that would not conduct heat so well? It seems to me that a solid slab would be as good as 8 in. of vermiculite.

MR. BOESTER: The house started out as a study for the *Vermiculite Research Institute,* the house using vermiculite, concrete, and light-weight steel frame. The house was built with only a floor, four walls and the roof. Later on, the research director for the *Vermiculite Research Institute* thought the house should be completely finished. The rest of the equipment was installed and at that time it was decided to put in a panel heating installation. The interest then became rather intense and a program of panel heating involving several systems was developed. The house, therefore, had the vermiculite concrete slab because of the study for another purpose.

W. L. MCGRATH, Syracuse, N. Y.: I would like to compare Fig. 1, showing the *on-off* thermostat control, and Fig. 3, showing the reset control. The inference is that the on-and-off thermostat is not suitable for such control. These two systems both have a single rate of firing. Therefore, the fineness of control evidenced in Fig. 3 must be due to more rapid cycles of the burner. This raises the question as to whether the use of an *on-and-off* thermostat has been fully explored. I notice on the chart that the cycles run one hour to one and a half hours, with the *on-and-off* thermostat, which seems abnormally long.

With a proper type of accelerating heater it is possible to reduce the operating cycle of the *on-and-off* thermostat to a much shorter period than is evidenced here. Therefore, I do not believe that a conclusion can be drawn at this time without qualifying the amount of artificial heater effect on the thermostat, if any were used.

The second point I would like to comment on is Professor Hutchinson's remark about the desirability of reducing air temperatures with increase in load and panel temperature. This can be done very simply with the modulating type thermostat, by judicious selection of the throttling range. A droop will occur from no load to maximum load and this effect can be deliberately used to depress the air temperature as the load increases. This is also true of an *on-and-off* thermostat using an anticipating heater where the amount of artificial heat can be chosen to depress the control point by the desired amount as the load increases.

FERDINAND JEHLE, Indianapolis, Ind.: When the temperature is controlled by room thermostat, why not let it operate the control valve or the three-way valve directly? Why have a bulb in the water stream? If we want the room air at a certain temperature, then why not directly let the thermostat operate the control valve?

There is one other thing I should like to point out. It is my understanding that only one room was under test. If five rooms had been under test, where would we locate the room thermostat? If we located it in the room, which happens to have sunshine at the time, I grant you that that room would probably be kept at very nearly the correct temperature. Four other rooms might at that time be too cold.

PROFESSOR HUTCHINSON: I think there is every reason to believe, as Mr. Jehle says, satisfactory results might have been obtained with a different arrangement. The reason we used only one room was that we wanted to have exact control of ventilation. You will notice a rate of one and a half air changes per hour was used, and in figuring the infiltration rate, we came up against the problem of measuring infiltration. We could not determine how much air entered through the cracks, so we sealed all the cracks and doors in the living room, with the exception of one opening under one door, and put a mechanical ventilation system in a separate room, to introduce air at a measured rate. Therefore, we were able to say with assurance that we had one and a half changes per hour because we introduced that amount of air.

MR. JEHLE: I did not point out *one room vs. all of them* because I thought Professor Hutchinson had overlooked this, but I wanted to make sure that that point was kept in mind.

R. G. VANDERWEIL, Woodbury, Conn.: The first question I would like to ask Professor Hutchinson is an old one. Are we really sure that the room air temperature is not a criterion of our comfort for all practical purposes even though this might not be so theoretical? In any case, let us look first at these test results, assuming that the room air temperature may be maintained constant. If this is the case, it seems as though the solution in Fig. 8*A*, Test No. 1 of this paper, is almost as favorable as the solution shown in Fig. 10, Test No. 3. The variation in temperature, that is, if we disregard the influence of the sunlight in Test No. 3, is practically zero, but also in Test No. 1 it does not seem to exceed 2 deg, or possibly 1 deg.

Now, if the room air temperature is the criterion for comfort I believe a 1 deg change of this temperature should not be critical. I still remember, when Professor Hutchinson went with our group to inspect the test house and in spite of our warning did not wear his overcoat. We then left a room heated to 70 F, with the outside temperature slightly below freezing. We exposed ourselves to a 40 F temperature drop and since we did not suffer under thermal shock I believe that 1 or 2 deg change of temperature should hardly influence room comfort, for practical purpose that is. Now, if we can maintain almost identical room temperatures with a very simple control unit, does the author feel that the use of the more complicated unit is warranted?

The next assumption is that for comfort the room air temperature cannot be considered alone, but only in connection with the rooms—MRT (mean radiant temperature). I realize that theorectically this must be true, but practically, only if the MRT changes considerably with the outside air. As Dr. Winslow mentioned, our body must lose heat at a certain rate in order to maintain comfort. That means that at all times the total of radiation and convection loss must remain constant (disregarding humidity).

Now, let us see how great a change of air temperature is required in order to conform to this principle. The author said that with low outside temperatures our panel surface temperature must be greater. But then, by the same token, the surface temperatures of the glass areas are considerably lower and that of the wall is lower too. Hence, if we want to give fair consideration to the desired change of the air temperature with the outside temperature, we would have to add one more line in Figs. 8*A* to 11, that is, the MRT of the room.

I believe with Professor Hutchinson, that the actual MRT of the room will vary with the outside temperature and that it will increase with decreasing outside temperatures. However, I am doubtful that this increase of MRT is so considerable as to call for a depression of the air temperature exceeding perhaps 2 F. In order to obtain this information, I think it would be desirable to show the MRT curve for all tests.

Only the addition of this curve may prove that a change of room air temperature with changing outside temperature is desirable. We may find that we have to change this temperature for 5 deg, and if this is so, we should build controls accordingly. However, if the depression is in the order of 1 deg, I think we can disregard it.

I would like to learn what differences will be found in the behavior of floor and ceiling system due to their different heat storage capacity. For practical reasons this seems to be an important item, and I have found much controversial data from various installations. I discussed this inertia problem a few weeks ago with engineers of one of our major control manufacturers and they told me that they checked one floor panel which had such great heat storage that once during the test period the heat went off and stayed off for 36 hours.

A few days later I happened to see a floor installation, which was controlled by outdoor-indoor thermostat. The owner, who practically lived with his radiant system for several weeks, told me that his room thermostat remained *up on the dot* during the entire period which included great variations in outside temperatures, and that the pump and boiler ran fairly continuously during this period.

So I wonder if we know enough about the heat lag and if you have further explanation or data explaining this problem.

Professor Hutchinson: This paper had nothing to do with comfort and therefore the MRT rise and air temperature relationship found in three *control* tests cannot properly be interpreted in terms of comfort.

Mr. Vanderweil: You mentioned the complicated inter-relations between air and MRT, due to the fact that as outside air and wall surface temperature drop, the panel temperature must be raised. Hence, it may seem that the simplest means of regulating a radiant heating system is still by room thermostat. In other words, before we know these complicated inter-relations which, by the way, will vary for each individual room, should we not still rely on the room air temperature?

Professor Hutchinson: The material in the paper represents experimental fact. As a research man, I prefer to leave to you the problem of interpretation.

Enrique Rumoneda, Mexico: Well, the only thing I have to say is panel heating is just in its first step in Mexico. I, particularly, have only finished a panel heating job for a restaurant, and I hope it will work well.

W. Bruce Morrison, Portland, Ore.: I think I can ask this question from the floor. I was wondering if that is a heat-anticipating type of thermostat that you used in the *on-off* control in your first test.

Professor Hutchinson: No, it was not.

R. D. Watt, Seattle, Wash.: In regard to thermostats, were the thermostats used standard commercial equipment? Was any attempt made to shield them? Where were they located? Have you done any experimenting with thermostats in the outside walls?

Professor Hutchinson: They were standard thermostats. Shielding was used, but only with respect to solar radiants. The thermostats were located at 5-ft air levels. No tests were made with the thermostat on the outside wall.

W. C. Schmitt, Rochester, N. Y.: The discussion seems to bog down mostly on residential heating. What happens when we go into industrial heating where we have considerable uninsulated surfaces, large glass areas and large masses of machines? Is there any advantage in saving, and comfort for the workers under those conditions, due to radiant heat?

Again, where you have such big expanses requiring much larger coils, do you have any trouble in imbedding them in the concrete, with heavy machinery on them?

Professor Hutchinson: I do not think that these questions are pertinent to this morning's discussion which is limited to control. Therefore I will not attempt to answer them.

Mr. Jehle: If I remember correctly, Mr. Vanderweil stated that the results in Fig. 8, etc., where the air temperature has not changed much, were close enough from the practical standpoint, and therefore, the control was good enough.

While I am not at all in favor of putting Professor Hutchinson's single equation on every single floor and ceiling panel, yet we cannot put a system in a man's house and ask him if he liked it, and base our conclusions entirely on that. In other words, what I am trying to get across is that the public has never demanded anything. They did not demand the steamship. They did not demand the automobile, and they did not demand panel heating. It was always one of us men who had a swell idea and sold it to them.

So, when we put in a control into a man's house, he may be satisfied with it, in fact might have been perfectly satisfied with a heating stove in the living room, but a competitor, if I may get commercial to that extent, can sell him the idea he ought to throw it out and put in a good one. I think it is up to us engineers to interpret the charts rather than the *practical* man, or the home owner.

Now, one remark about Professor Hutchinson going out in the cold. What man does by choice he finds no fault with. If we compelled him to do it, I am sure he would put on an overcoat or ask for one.

MR. VANDERWEIL: Just a word in answer to Mr. Jehle. This was one of the questions I asked Professor Hutchinson, but it remained unanswered. Should we use most simple or complicated regulating equipment? I could not find the answer, and maybe Mr. Jehle can give it to us.

When controlling room temperatures, a second item, psychology, seems to be of greater importance than comfort.

I agree with Mr. Jehle that Professor Hutchinson felt *comfortable* only because he voluntarily did not use his coat. Also, in installations I have seen occupants voluntarily set their room thermostat to 70 F, or 1 to 2 deg above and operate at this position throughout the heating season. However, in radiant heating installations devoid of room thermometers, the control will invariably operate at temperatures slightly below 70 F, sometimes at 65 F. Apparently then, the psychological moment outweighs the effects of small temperature changes upon physical comfort, which again indicates that a 2 F change of room temperature is not of great importance.

1312

No. 1312

PROPOSED DESIGN PROCEDURE FOR LARGE MECHANICAL WARM AIR HEATING SYSTEMS

By S. Konzo*, R. J. Martin**, D. S. Levinson†, and R. W. Roose‡, Urbana, Ill.

In 1940, the Installation Codes Committee of the *National Warm Air Heating and Air Conditioning Association* published a Design Manual No. 7, entitled *Code and Manual for Design and Installation of Warm Air Winter Air Conditioning Systems.* This Manual was intended for smaller structures having design heat losses less than 120,000 Btu per hour, and for furnace-blower combinations having the following performance characteristics:

1. Temperature rise through furnace not to exceed 100 F.
2. Static pressure available for overcoming pressure losses in distribution system of 0.20 in. water gage.

A demand was expressed by industry for a similar design manual applicable to structures having a design heat loss greater than 120,000 Btu per hour, and in which a wide range of values for temperature rise, static pressures, and other variables would be offered in place of the fixed values given in *N.W.A.H.A.C.A. Manual No. 7.* This study should be considered as an exploratory outline of possible design procedure and should indicate desired areas of additional research. It is hoped that suggestions will be offered by industry which will aid the Codes Committee in formulating a workable procedure that will be sufficiently simple for the heating engineers to handle.

In any overall survey of this nature, no single source of reliable reference material is available. The work has consisted primarily of a systematic and thorough search of data from every available source. In this connection, liberal use has been made of data from A.S.H.V.E. publications, particularly those

*Special Research Professor, University of Illinois. Member of A.S.H.V.E.

**Assistant Professor—University of Illinois.

†Graduate Student, University of Illinois. Student Member of A.S.H.V.E.

‡Special Research Assistant, University of Illinois. Junior Member of A.S.H.V.E.

Presented at the 53rd Annual Meeting of the American Society of Heating and Ventilating Engineers, Joint Session with the *National Warm Air Heating and Air Conditioning Association,* Cleveland, Ohio, January 1947.

issued under the auspices of the Research Technical Advisory Committee on Air Distribution and Air Friction and the Committee on Code for Testing and Rating Heavy-Duty Furnaces.

PRELIMINARY PROCEDURE

In general terms the proposed design procedure sets forth the following preliminary steps:

1. Calculation of design heat losses from individual spaces in the structure. This calculation is made by commonly accepted methods, such as stated in the HEATING, VENTILATING, AIR CONDITIONING GUIDE.

2. Drawing of floor plans of structure showing proposed location of registers and return intakes. The type of register, whether wall or ceiling type, and the desired throws and deflections, are to be specified.

3. Laying out a proposed duct system for both the warm air and return air sides of the system. This includes details of types of fittings in the trunk line and in each branch line, and the actual and equivalent lengths of each branch line, from bonnet to register. It does not, however, include sizes of ducts or registers which are still to be determined.

DETERMINATION OF AIR VOLUME AT REGISTER

The volume of heated air required at a warm air register, to offset the heat loss from a space, is given in an equation of the form:

$$Q = \frac{H}{(60)\,(0.242)\,(d)\,(t_r - t_o)} \quad \text{(1)}$$

in which

Q = quantity of air, at density corresponding to register air temperature, in cubic feet per minute.

H = hourly heat loss of space at design condition, including transmission and air infiltration losses, in Btu per hour.

d = density of air corresponding to register air temperature, pounds per cubic foot.

t_r = register air temperature, Fahrenheit.

t_o = temperature at return intake, Fahrenheit.

For a given register air temperature, t_r, and for an assumed value of 65 F for the return air temperature t_o, Equation 1 can be expressed in the form:

$$Q = H \times \text{Factor} \quad \text{(2)}$$

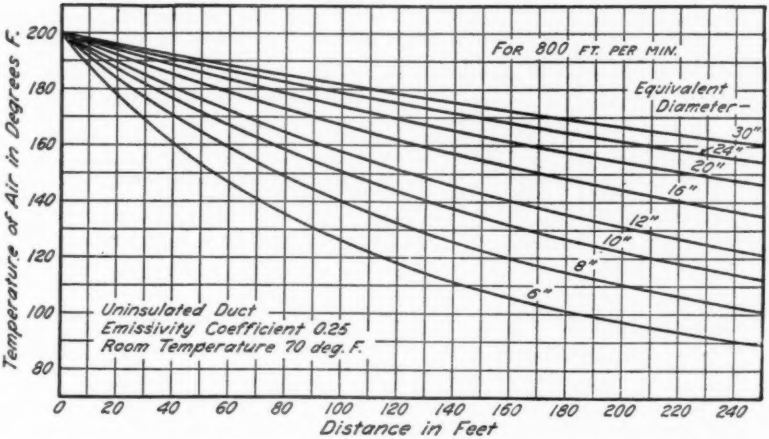
in which the factor is a function of the register air temperature alone. For example, for a register air temperature of 140 F, the factor is 0.014 for 1 Btu per hour, or 14 for 1000 Btu per hour loss. Therefore, if the register air temperature can be determined, the air volume requirement can then be established for any space for which the hourly heat loss is known.

SELECTION OF BONNET DESIGN TEMPERATURES

The temperature of the air leaving any register is dependent upon a large number of variables, including temperature of air at entrance to duct, size and length of duct, velocity of air stream, and temperature of ambient air and radiat-

ing surfaces. For the purposes of this design manual, data submitted by Kratz, Konzo, and Engdahl [1], as shown in Fig. 1, were utilized. Over a range of bonnet air temperatures from 110 F to 200 F the approximate register air temperatures for varying lengths of ducts, and the corresponding cfm requirements per 1000 Btu per hour loss, were determined, as shown in Fig. 2.

In deriving the values in Fig. 2, the assumption was made that the average size of duct was 10-in. in diameter, and that the air velocity was 800 fpm. The register air temperatures, shown in the upper part of each square in Fig. 2, would be slightly high for velocities less than 800 fpm and for ducts smaller than



(Reproduced from University of Illinois Engineering Experiment Station Bulletin 351)

FIG. 1. TEMPERATURE DROP IN DUCTS FOR AIR VELOCITY OF 800 FPM.

10 in. in diameter, and hence would give lower values of cfm requirements for 1000 Btu per hour loss than if corrected values were used. However, calculations made for a 6 in. duct carrying air at 400 fpm velocity, indicated that the maximum discrepancy was only of the order of 6 percent. Conversely, the cfm requirements shown in Fig. 2 would be slightly on the safe side for ducts larger than 10 in. in diameter and for air velocities greater than 800 fpm. Hence, in spite of the fact that the values in Fig. 2 were based upon a specific diameter and velocity, the results obtained from their use should be well within the accuracy required for ordinary calculations.

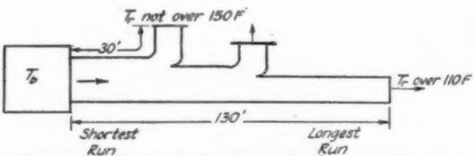
The Committee on A.S.H.V.E. Code for Testing and Rating Heavy-Duty Furnaces has under consideration the statement of capacity ratings of the furnace for three or more specific rises of temperature, from blower inlet to bonnet of the furnace. It is possible for the installer of such equipment to arbitrarily

[1]Temperature Drop in Ducts for Forced-Air Heating Systems, by A. P. Kratz, S. Konzo, and R. B. Engdahl. (University of Illinois Engineering Experiment Station, *Bulletin* 351, 1944.)

select one of the temperature rises, and therefore to specify the bonnet temperature to be used for design purposes.

However, for those installations in which a bonnet temperature is not specified, it is proposed that Fig. 2 be used for the selection of a design bonnet temperature. If, under normal conditions, the maximum and minimum register air temperatures for a given installation are specified as approximately 150 F and 110 F, respectively, it may be noted from the broken lines in Fig. 2 that the selection of a design bonnet temperature can be made as follows:

1. Using the longest actual length, from bonnet to register, follow the vertical column downward until the broken line is crossed. Read the minimum permissible tempera-



Bonnet Temp.	Length in feet (Bonnet to Register										
	10	20	30	40	50		120	130	140	150	etc.
110											
120											
130								105.4 24.6			
140							112.2 21.1	110.4 21.7	109.0 22.5		
150			140.0 14.0		134.0 15.1	T_r →		116.0 19.6			
160		132.4 12.3	148.7 12.7	145.2 13.2		C.f.m. / 1000 Btu per hr					
170 etc.			157.2 11.7								

Design bonnet temp. can be 140 F, 150 F, or 160 F. Use 150 F.

FIG. 2. PROPOSED TABLE SHOWING RELATIONSHIPS OF BONNET TEMPERATURE, REGISTER TEMPERATURE, AND CFM REQUIREMENTS.

ture for bonnet air in the left-hand column. For example, for an actual length of 130 ft, the corresponding minimum bonnet temperature is 140 F.

2. Using the shortest actual length, from bonnet to register, in the same installation, follow the column downward until the lower broken line is reached. Read the maximum design bonnet temperature in the left-hand column. For example, for an actual length of 30 ft, the corresponding maximum bonnet temperature is 160 F.

3. Select any design temperature within the range thus determined. For the example given, a mean temperature of 150 F could be specified, although either 160 F or 140 F could also be used.

4. Regardless of whether the design bonnet temperature is determined arbitrarily or by the preceding method, once selected it must be used in all later calculations. The cfm requirements per 1000 Btu per hour loss can then be readily determined from the lower of the two numbers in each square in Fig. 2. For example, in the installation under consideration, a design bonnet temperature of 150 F can be assumed. For actual lengths from bonnet to register of 30, 50, and 130 ft, the corresponding cfm require-

ments are 14.0, 15.1, and 19.6, respectively, for each 1000 Btu per hour heat loss. The procedure given permits the designer a wide latitude in the selection of a bonnet temperature. At the same time, some consideration is given to the heat loss from ducts, so that long branch runs are supplied with relatively larger quantities of heated air than are short runs.

Selection of Blower and Furnace

At this point in the design procedure the following information is available, which will be of assistance in the selection of the blower and furnace:

1. *Cfm Air Requirements at each Register.* The air volume requirements at each register are separately determined by the use of Equation 2. The volumes thus obtained are for an air density corresponding to the temperature at each register, and can be corrected to conditions of standard density of air of 0.075 lb per cu ft. A summation of these corrected volumes can be considered as the air delivery requirement of the blower, for which the ratings are usually expressed in terms of standard density.

A simpler approach would be to make a summation of the actual cfm air requirements, as obtained from Equation 2, and arbitrarily to assume that this sum is the volume at standard density to be handled by the blower. Over a range of average register air temperatures of from 110 F to 150 F, this procedure will give cfm totals about 8 to 15 percent greater than those determined by the exact method. However, in view of the fact that some amount of air leakage will occur in the duct system between the blower and the registers, the use of the larger cfm values could be justified, and is therefore recommended.

2. *Temperature Rise through the Furnace.* With the aid of Fig. 2, the design value of the bonnet temperature was previously determined. The difference in temperature of the air at the bonnet and that entering the blower represents the temperature rise through the unit. In the case of heavy-duty furnaces, ratings based on three specific temperature rises of 50, 75, and 100 deg F are under consideration. From the standpoint of the heating contractor, rating curves covering a wide range of temperature rises would be more convenient to use, and would permit a closer selection of blower capacity.

3. *Heat Delivery at Registers, and Bonnet Capacity of Furnace.* The summation of the hourly Btu losses from the rooms or spaces in the structure to be heated is equal in magnitude to the heat delivery at the registers. Furthermore, the required bonnet capacity can be determined from the air volume requirement, (as calculated from the preceding step) and the temperature rise through the furnace.

In the case of furnaces to be used in smaller structures (N.W.A.H.A.C.A. Manual No. 7), an allowance of 15 percent was made to take care of the heat loss of the duct between the bonnet and the registers. That is, when the register delivery was obtained, the required bonnet capacity was determined by dividing the Btu delivery by 0.85. However, in the case of heavy-duty furnace installations, the average length of ducts will vary considerably from one installation to the next, with the result that the heat loss from the ducts will also vary widely. Hence, it is not feasible to state a fixed relationship between register delivery and bonnet capacity.

The location of the furnace in relation to the space to be heated will determine whether the required bonnet capacity or the required register delivery is the significant value to be used in the selection of a furnace. For those installations in which the furnace is used as a *unit heater,* and is located in the space to be heated, any duct heat loss may be considered as useful heat, and the required bonnet capacity could be considered as equal in magnitude to the required register delivery. However, for those installations in which the furnace is located in a separate heater room, only a portion of the duct loss will be regained, and the bonnet capacity alone is considered as the significant item to be used in selecting the furnace. In general, the difference between

the bonnet capacity and the register delivery represents the *piping loss* of the system, and may also be considered as representing some amount of reserve capacity.

For those cases in which the equipment is used for buildings to be heated intermittently, some reserve capacity may be needed for the pick-up load. As yet, no decision has been made by the heavy-duty furnace committee as to the desired magnitude of this reserve capacity. Furthermore, no conclusion has been reached as to whether the duct system should be designed for the normal load or the pick-up load. If the latter procedure is decided upon, it may necessitate a revision of the original cfm requirement at each register.

It is obvious that the decisions made by the heavy-duty furnace committee will affect the final procedure for design of the duct system and selection of the equipment. It is hoped that some uniformity in practice will eventually be obtained.

SELECTION OF REGISTER SIZES AND PRESSURE LOSS.

The problem that faces the designer at this point is complex, in that a number of interrelated items must be considered separately and concurrently. Two cases can be presented that illustrate the point:

Case 1: A factory assembled heating unit, consisting of a furnace and blower, has a rated bonnet capacity of 300,000 Btu per hour and is equipped with a blower having a rated delivery of 4000 cfm at 0.75 in. static pressure. Assuming that the pressure loss through blower, cabinet, furnace casing, and filters is 0.35 in., the available static pressure for the return and supply duct system is 0.75-0.35, or 0.40 in. If a further assumption is made that the pressure available in the bonnet is 0.20 in., leaving 0.20 in. for the loss in the return side, it is apparent that the total loss of the ducts, fittings, and registers cannot exceed 0.20 in. For those installations in which relatively long throws of air are required at the register, an available pressure behind the register of 0.20 in. or more would be frequently encountered. In such cases, the 0.20 in. pressure available at the bonnet is not sufficient to take care of both register and duct work. Obviously, a better procedure would be first to determine the register loss, and then to decide what pressure is available for the ducts.

Case 2: A designer has the option of selecting a blower independently of the furnace. For example, a furnace with a rated bonnet capacity of 300,000 Btu per hour can be installed with a blower having a rated delivery of 4000 cfm at 0.75 in., or one having a delivery of 4000 cfm at 1.0 in. The latter blower would probably be larger in size, greater in initial cost, and more expensive to operate than the former. On the other hand, a greater allowable pressure would permit the use of high-pressure registers having long throws, as well as smaller-sized duct work. In this case, also before any selection of blower or duct friction loss can be made, some evaluation of pressure loss at the register is necessary as a first step.

A thorough search of a number of references, including practically all manufacturers' catalogues, indicated the need of a summary table embodying complete performance characteristics of a wide range of register sizes. Wherever such data for a specified model are stated in the catalogue, they are preferred to any typical results that may be derived for an *average* register. Nevertheless, the discrepancies that were noted in the data from various sources, as well as partially complete listings in many sources of material, led to the formulation of a table, which will be explained and shown later.

This table, when completed, will show a summary of the best available data, from a number of sources, for registers of commercial types having horizontal or vertical bars, and for a zero degree deflection angle of the air stream. The table does not apply to perforated grille type faces. Similar tables are proposed, but not shown, for bar-type registers having angles of deflection of 22 deg and 30 deg.

For a given throw and air volume to be handled, the following information can be obtained: (*a*) register free areas, square inches; (*b*) pressure loss, inches of water; (*c*) sound level, decibels. In addition, information relative to (*d*) selection of nominal register size and (*e*) effect of deflection angles on pressure loss can also be obtained.

(*a*) *Relation of Area to Throw:* The data presented by Tuve and Priester[2] were selected as the most comprehensive found in a number of references that were investigated. The basic equation that they proposed was of the form:

$$V_r = \frac{KQ}{X\sqrt{A_e}} \qquad (3)$$

in which

V_r = residual maximum velocity, feet per minute.
K = constant.
Q = volume of air discharged, cubic feet per minute.
X = throw of air stream, feet.
A_e = effective area of outlet, square feet.

Furthermore,

$$A_c = A_e/C \qquad (4)$$

and

$$V_c = Q/A_c, \text{ or } A_c = Q/V_c \qquad (5)$$

In which

A_c = core area of register, square feet
C = coefficient of discharge
V_c = core velocity, feet per minute

By rearranging Equations 3, 4, and 5, the following form was derived:

$$V_c = \frac{X^2}{Q}\left(\frac{CV_r^2}{K^2}\right) \qquad (6)$$

Numerical values for V_r = 50 fpm, K = 2.5, and C = 0.61 were inserted in Equation 6 and the relationship was simplified to the form:

$$V_c = \frac{X^2\,(244)}{Q} \qquad (7)$$

By substituting numerical values for X and Q in Equation 7, values of V_c were obtained, and these in turn were substituted in Equation 5 to give the required core areas, A_c.

In order to obtain free areas, which are usually stated in manufacturers' catalogues, the assumption was made, after consultation with the register manufacturers, that the ratio of free area to core area was 0.78. Hence, by the use of a modified version of the Tuve-Priester equation (Equation 3), relationships between throw, volume, and free areas of registers could be obtained.

In the opinion of some plant designers who have been consulted, a residual velocity of 50 fpm at the wall opposite a register was considered as too large. In their opinion, average stream velocities of the order of 15 to 20 fpm were considered as more reasonable. It is possible to modify Equation 7 by inserting, for instance, a value of 15

[2]Control of Air Streams in Large Spaces, by G. L. Tuve and G. B. Priester. (A.S.H.V.E. TRANSACTIONS, 1944, p. 153.)

fpm for the residual velocity. However, it is questionable whether such an extrapolation to velocities which are practically unmeasurable can be justified. The suggestion has been made that the length of *throw* should be considered as equal to, say, three-fourths of the width of the space, instead of the entire width, and that the values obtained from Equation 7 be applied to this shortened distance. Since no definite research results are available at the present time, it is probable that the use of this arbitrary and inadequate rule may be necessary as a temporary expedient.

(*b*) *Relation of Velocity to Pressure Loss.* The pressure loss through a register can be expressed in terms of (*a*) inches of water, (*b*) velocity head loss, or (*c*)

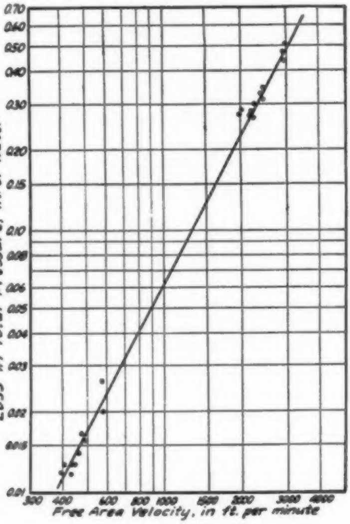


FIG. 3. RELATIONSHIP BETWEEN FREE-AREA VELOCITY OF REGISTER AND TOTAL PRESSURE LOSS.

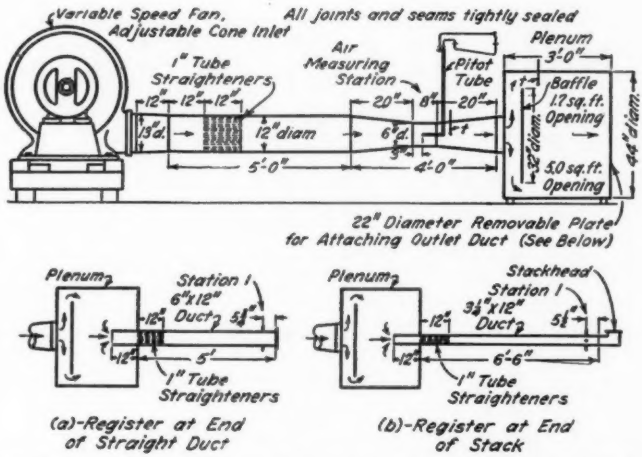
equivalent feet of duct leading to the register. From the standpoint of the designer, pressure losses stated in terms of equivalent feet of duct are most convenient to use, since resistance of fittings can be similarly expressed. However, in the cases of large installations no definite relationship exists between the size of the register and the size of the duct leading to it. Hence, no feasible method could be devised for expressing pressure losses of registers in terms of equivalent feet of duct. Neither did an expression of pressure losses in terms of velocity heads offer any simplification of approach. Hence the decision was made to express losses for registers in terms of inches of water.

For this relationship, data from Kratz and Konzo[3] were used. Fig. 3 shows the relation between the free area velocity and total pressure head of the register for 23

[3]Pressure Losses in Registers and Stackheads in Forced Warm Air Heating, by A. P. Kratz and S. Konzo. (University of Illinois, Engineering Experiment Station, *Bulletin* 342, 1942. Data shown in Fig. 4 were plotted from data in Table 2, page 115, from the same reference.)

types of commercial models of widely varying design, in which no air deflection was used. These tests were conducted in a plant shown in Fig. 4. The total pressure ahead of the register included not only the losses due to friction, turbulence, and expansion, but also the velocity head of the air leaving the register.

(*c*) *Sound Level:* The work of Stewart and Drake[4] on sound level of registers was selected from a number of references consulted. A communication from D. J. Stewart recommended that, instead of the figure now in current use,[5] which is for horizontal fin type registers only, a new curve be drawn based on averages of results obtained by the same authors, for registers with horizontal and vertical fins.[6] The



(Reproduced from University of Illinois Engineering Experiment Station Bulletin 342)

FIG. 4. DIAGRAM OF LABORATORY PLANT FOR MEASUREMENT OF PRESSURE LOSSES IN REGISTERS AND FITTINGS.

curves thus obtained are shown in Fig. 5, and values read from the curves were used to locate the diagonal broken lines in Fig. 6.

(*d*) *Relation of Free Area to Nominal Register Size:* It is realized that some differences exist in the ratio of free area to the nominal area of registers, especially in the case of perforated grille types. However, a survey of a number of manufacturers' catalogues indicated that the divergence in the ratio of free area to nominal area was not too great in the case of fin or bar type registers, and certainly not great enough to preclude the establishment of a register selection chart, as given in Fig. 7. The nominal sizes of registers indicated in this figure may be subject to some changes. However, they do represent the sizes which a number of manufacturers have indicated as being desirable.

(*e*) *Effect of Deflection Angles on Pressure Loss:* Tables similar to Fig. 6 are proposed for registers having deflection angles of 22 deg and 30 deg. For those

[4]The Noise Characteristics of Air Supply Outlets, by D. J. Stewart and G. F. Drake. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 81.)

[5]Air Flow and Loudness Chart. HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1946, p. 773, Fig. 5.

[6]Material furnished by D. J. Stewart.

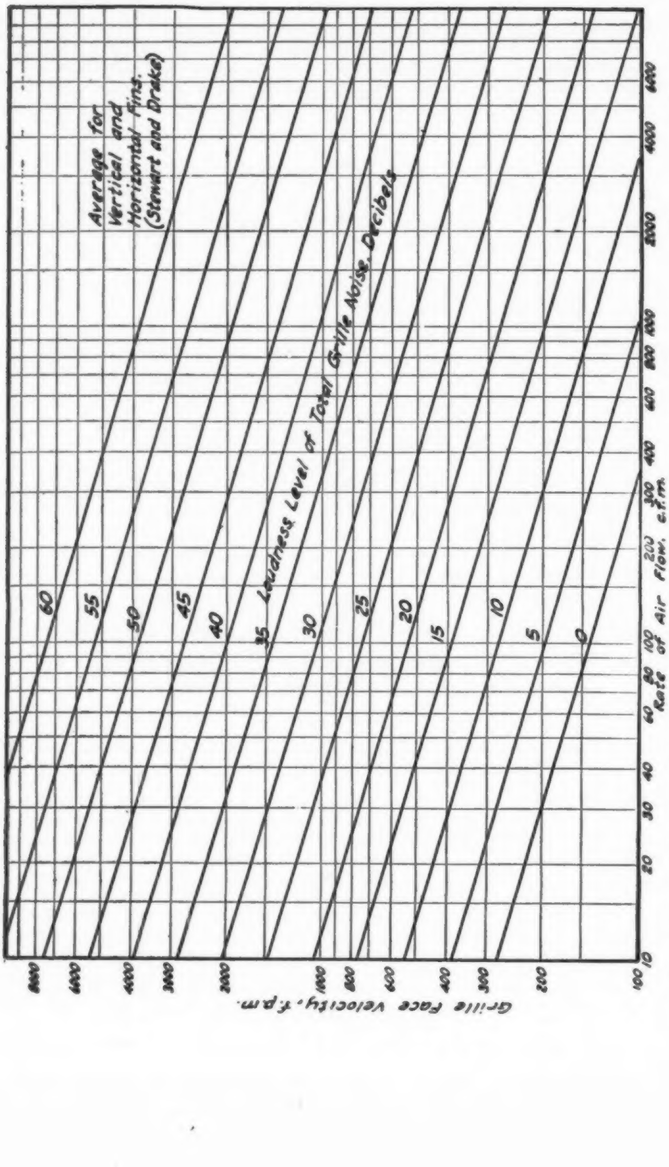
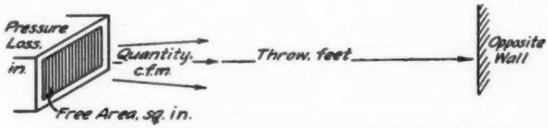


FIG. 5. LOUDNESS LEVELS FOR REGISTERS. (AVERAGE VALUES FOR VERTICAL AND HORIZONTAL FINS. DATA FROM STEWART AND DRAKE.)

portions of the table referring to pressure loss, the relationships[7] shown in Fig. 8 will be used. The data were obtained from eight different registers tested in the plant shown in Fig. 4. It is apparent that the pressure loss increased relatively slowly when the angle of deflection was increased from 0 deg to about 15 deg, but increased at a greatly accelerated rate for angles larger than 15 deg.

In many respects, Fig. 6 represents an audacious move to generalize data that are inherently not subject to generalization. The data are offered as practical expedients which may aid the designer, until the time is reached when a majority of manufacturers will be able to offer similar data for their specific models.



Quantity, c.f.m.	Throw, ft.						
	10-12	13-15	16-18	19-21	22-24	25-29	etc.
60-99	24 0.02	15 0.04	10 0.08	8 0.14	7 0.24	← Free Area ← Loss	
100-139	55 0.01	34 0.02	23 0.04	17 0.07	13 0.11	9 0.21	
140-179		60 0.01	41 0.02	30 0.04	22 0.06	16 0.12	B
180-219		95 0.01	59 0.01	47 0.03	35 0.04	25 0.08	35 decibel
220-259			92 0.01	66 0.02	50 0.03	36 0.06	limit
etc.			etc.		A		
				20	decibel limit		

REGISTER THROW, FREE AREA, AND LOSS.

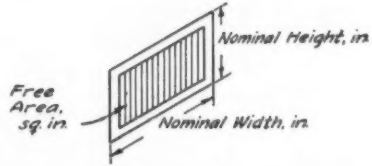
FIG. 6. PROPOSED TABLE GIVING RELATIONSHIPS OF AIR THROW, FREE-AREA, PRESSURE LOSS, AND SOUND LEVEL FOR REGISTERS (FOR REGISTERS OF THE NON-DEFLECTING TYPE).

SELECTION OF DESIGN STATIC PRESSURES FOR DUCT SYSTEM

The pressure loss of each register in the duct system can be obtained by the use of data similar to those shown in Fig. 6. Furthermore, as discussed in a preceding section, entitled *Selection of Register Sizes and Pressure Loss,* the available pressure in the bonnet will be specified or assumed. In any case, it is apparent that the bonnet pressure will have to be greater than the maximum pressure loss of any register in the system. The differences between the bonnet pressure and the pressure loss of each register will represent the available pressure loss for the duct system connecting the bonnet to the register.

For example, as shown in Fig. 9, if the pressure loss of register (*a*) is 0.12 in., for register (*b*) is 0.01 in., and for register (*c*) is 0.03 in., then the maximum pressure loss is 0.12 in. The required bonnet pressure must be greater than 0.12 in. For example, if the bonnet pressure of the unit selected

[7]Pressure Losses in Registers and Stackheads in Forced Warm Air Heating, by A. P. Kratz and S. Konzo. (University of Illinois Engineering Experiment Station, *Bulletin* 342, 1942, p. 31.)



Free Area, sq. in.	Nominal Register Size, in.					
18 or less	8×4					
19 - 26	8×5	10×4				
27-32	8×6	10×5	12×4			
33-42		10×6	12×5	14×4	16×4	
43-52		10×8	12×6	14×5	16×5	18×4	20×4
etc.			etc.			

FIG. 7. PROPOSED REGISTER SELECTION TABLE.

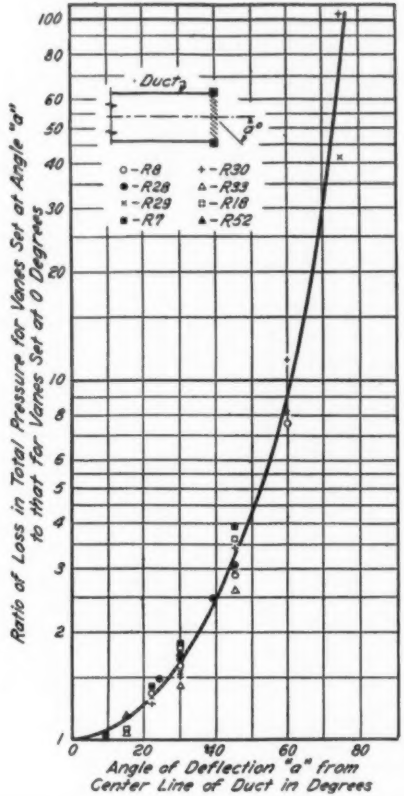


FIG. 8. *(Right)* INCREASE IN PRESSURE LOSS WITH DIFFERENT ANGLES OF DEFLECTION.

(Reproduced from University of Illinois Engineering Experiment Station Bulletin 342)

is 0.25 in., then the available pressure for overcoming resistance in each of the runs from the bonnet to the register will be for (*a*) 0.13 in., for (*b*) 0.24 in., and for (*c*) 0.22 in.

Once the total pressure loss for each run has been determined, the selection of the friction loss per 100 ft of duct for each run can be determined by dividing the available pressure for each run by the respective equivalent lengths of ducts from bonnet to register. The design procedure from this point forward is no different from that of a commonly used method, frequently referred to as the *equal friction pressure loss method*. For each branch duct in the system, both the cfm to be handled and the friction loss per 100 ft will be known. By the use of the common friction chart for air flow, the size of the ducts can be readily determined. A few refinements are proposed in the new *N.W.A.H.A.C.A.* Manual that will simplify the calculation required. For example, a table will be shown instead of the friction chart, and a separate table will be given for the relationship between round pipe sizes and equivalent sizes of rectangular ducts. Furthermore, a complete work sheet, which will enable a designer to follow a step-by-step procedure, is in preparation. Hence, by the use of the prepared tables and work sheets, it is the intent of the Installation Codes Committee to furnish the designer and the installer with working tools that will enable them to design a duct system for large, mechanical warm-air systems.

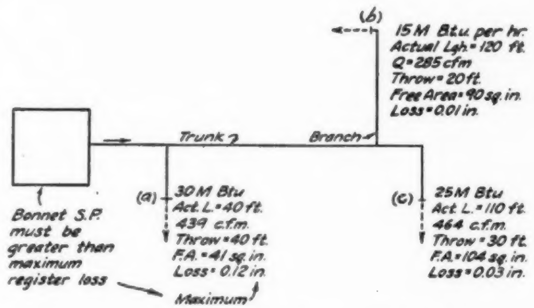


FIG. 9. MAXIMUM REGISTER LOSS AND ITS RELATION TO DUCT LOSS AND BONNET STATIC PRESSURE.

In this brief review, it is not possible to show in full detail all the proposed tables, nor to discuss in detail each of the many factors that were considered. Some important omissions have been made, but have not been overlooked. The authors are cognizant of the necessity of taking into account such items as: relation of ventilation load to both cfm requirements at register and furnace capacity; duct design to take care of both cooling load and heating load; magnitude of pressure loss through heavy-duty furnace units; a procedure for the selection of ceiling diffuser type outlets; effect of insulation of ducts on cfm

requirements; and many others. In view of the fact that the proposed procedure is still in the draft stage, any suggestions relative to the problem will be gratefully received.

DISCUSSION

H. B. NOTTAGE, Cleveland, Ohio: The need brought out in this paper for useful design data on the behavior of heated air jets should provide a valuable stimulus to activities being planned at the A.S.H.V.E. Research Laboratory and Case Institute of Technology. A program is currently being formulated to comprise some very interesting and fundamental work on this problem, which will be both experimental and analytical in nature.

We would be interested in hearing from anyone engaged in design and application problems, regarding their practical requirements for data of these types. Comments should aid us in making such data available quickly in the most useful form.

R. W. ROOSE, Urbana, Ill.: Equation 7 was revised to $V_c = \frac{X^2(320)}{Q}$ in the final form when the tables similar to Fig. 6 were developed for the new N.W.A.H.A.C.A. Manual. The change was brought about by using a value for $C = 0.80$ in place of 0.61, all other factors remaining the same. For this same table, the length of throw was taken as three-fourths the distance from the register to the opposite wall, with the terminal velocity of the air stream reaching 50 fpm at the end of the throw.

1313

No. 1313

DETERMINING AND REDUCING THE CONCENTRATION OF AIR-BORNE MICRO-ORGANISMS

By Matthew Luckiesh* and A. H. Taylor**, Cleveland, Ohio

THERE is abundant clinical evidence that some infectious and contagious diseases are spread by air-borne bacteria or viruses[1]. These organisms may be present in any room or building occupied by persons or animals. One obvious purpose of mechanical ventilation is the reduction of the concentration of such bacteria or viruses, in order to reduce the spread of disease. However, there are practical limits to the number of air-changes per hour which can be achieved satisfactorily by mechanical ventilation. Obviously, in addition to ventilation, some methods of destroying the air-borne organisms can be introduced. One very effective agent is radiant energy, especially that in the middle ultraviolet region, $\lambda 2000$ to $\lambda 3000$. The region of maximum effectiveness appears to be at approximately $\lambda 2500$-$\lambda 2700$.[2] As seen in Fig. 1, the lethal effectiveness. decreases rapidly as the energy increases in wavelength. Notwithstanding this fact, sunlight, which has practically no energy shorter than $\lambda 3000$, is a germicidal agent, largely because of the massive energy values and long exposures involved. However, very low wattages of socalled germicidal lamps (low-pressure mercury arcs) outdo sunlight in germicidal effectiveness.

Sources and Types of Bacteria

Many of the air-borne bacteria in occupied interiors have their origin in the respiratory systems of the occupants, or may be dislodged from their clothing, bodies, etc. Dust floating in the air or stirred up from the floor carries many bacteria. Generally a large percentage of those collected from the air are harmless to human beings in the concentrations usually found, but commonly some are pathogenic.

---

*Director, Lighting Research Laboratory, General Electric Co., Nela Park.
**Physicist, Lighting Research Laboratory, General Electric Co., Nela Park.
[1]Exponent numerals refer to References.
Presented at the 53rd Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January 1947.

The positive identification of colonies developed from bacteria collected from the air is a complex and tedious undertaking. In some field work by this laboratory, the air was sampled in the reception room of a doctor's office while occupied by 5 to 15 patients. One petri dish, containing 108 colonies, was selected for a detailed study of types collected, and 27 different types were

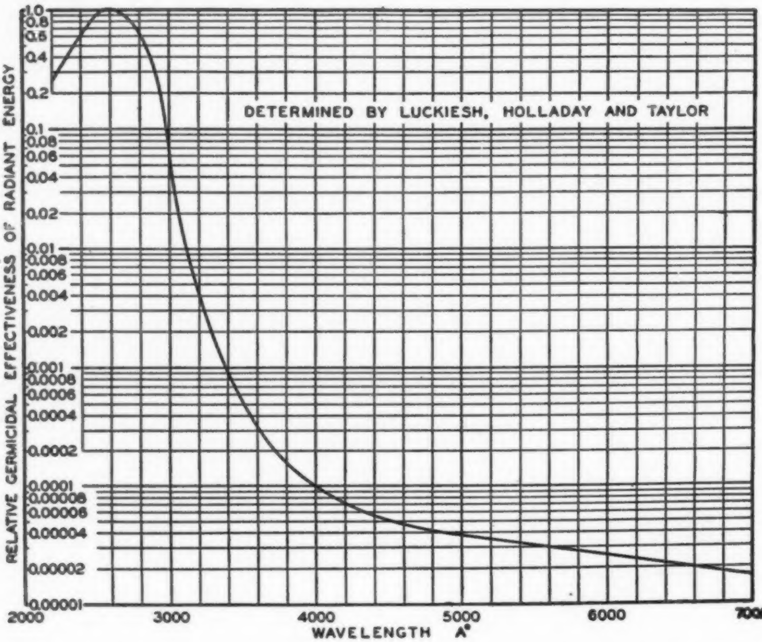


FIG. 1. RELATIVE GERMICIDAL EFFECTIVENESS OF RADIANT ENERGY OF VARIOUS WAVELENGTHS AS DETERMINED WITH B. COLI SEEDED ON AGAR PLATES AND ALSO IN SHALLOW DISHES OF DISTILLED WATER.

identified. Approximately half were of possible human origin, and approximately 20 percent of the total were potentially pathogenic.

Many micro-organisms are of such size that they can be observed in a high-power microscope. The rate of reproduction is so high that a single organism on a suitable culture medium may multiply, in less than 24 hrs, to such a large number as to be visible without magnification. However, some types must be incubated for several days, and even weeks, before becoming visible.

Viruses are much smaller than bacteria and cannot be seen in the highest power visual microscopes, but some types have been observed with the electron microscope. Some believe that they are living organisms, while others believe

they are autocatalytic bodies capable of producing injurious action. Regardless of their physical nature, they can be destroyed by suitable exposures to ultraviolet energy.[3]

COLLECTING AIR-BORNE MICRO-ORGANISMS

Two general methods of collecting air-borne bacteria and determining their concentration are in use.[4] By passing the air through water containing a nutrient material, then *plating out* a small amount of the water onto a culture medium and incubating it, the number of micro-organisms in a measured quantity of air may be determined. Several types of air-samplers collect the bacteria directly on a culture medium, which can then be incubated

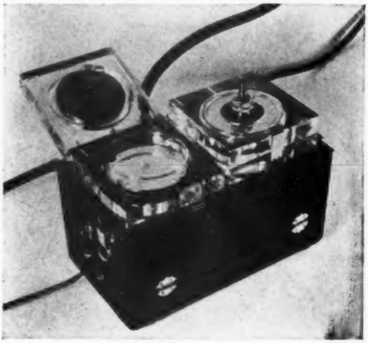


FIG. 2. EFFICIENT TYPE OF RADIAL-JET AIR-SAMPLER WITH TWO IDENTICAL SAMPLING UNITS, USEFUL IN MAKING COMPARATIVE STUDIES.

for growth of the colonies. Probably the first method of air-sampling was ***sedimentation,*** in which open petri dishes containing a suitable nutrient were exposed in the area to be studied. The bacteria collected by this method are principally those carried on the heavier particles of dust and the larger moisture droplets which are least supported by upward air drafts. Obviously this method does not yield directly quantitative values of the number of organisms in a given volume of air.

Several proposed air-samplers, notably the Wells Air Centrifuge, the Hollaender Funnel-Type Aeroscope, and the Bourdillon Jet Sampler employ the principle of impingement of the air on the culture medium.[4]

The authors and their colleague, L. L. Holladay, have devised a very efficient type of sampler employing the high-velocity impingement principle[5,6,7]. One of these, designated as a *radial-jet sampler* is illustrated in Fig. 2. Only a single unit is actually necessary, but in this illustration two identical units are employed for laboratory use to study the effect produced by various variable

factors such as intensity of germicidal energy, duration of exposure, humidity, etc. The petri dish containing a moist gelatin-like nutrient is placed on a metal platform, which is rotated slowly by a Telechron motor. One open unit shows the petri dish in position and the closed drum, with narrow radial slit, in the lid. At an air rate of 1 cfm, air issues from the slit with a linear velocity of approximately 100 fps, thus forcefully impinging the micro-organisms on the moist culture medium. The result is a very efficient air-sampler. However, its usefulness for air-sampling outside the laboratory is limited by the need for an air pump which is necessarily heavy.

The authors have also devoted a great deal of research to the development of an air-sampler employing the electrostatic principle of impingement of air-borne micro-organisms. It was soon found that a strong electrostatic field

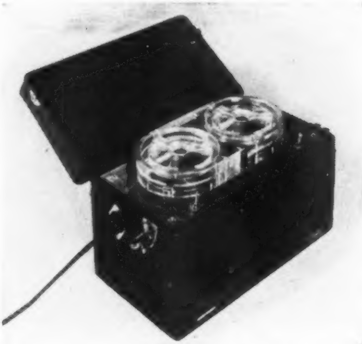


Fig. 3. Portable Duplex Electrostatic Air-Sampler Which Uses Standard Petri Dishes for the Collection of Air-Borne Bacteria.

greatly increased the efficiency of collection. A result of these researches is a portable sampler weighing less than 15 lb, illustrated in Fig. 3. Air enters each unit of the sampler at the center of a 150 deg metal cone connected to one side of a high d-c voltage, the petri dish with nutrient being placed on a metal plate connected to the other side of the high voltage. Since the carriers of bacteria in the air may be charged either positively or negatively, it is necessary to provide for the collection of both types of charged particles. This is accomplished by having two identical sampling units, in which one unit has the cone positive and the lower plate negative, while the reverse polarity is used in the other unit. Air is drawn through each unit at ½ cfm by means of a small blower located in the housing below the sampler units. Thus, if the sampling period is 10 min, one unit collects the positively charged particles in 5 cu ft of air and the other unit collects the negatively charged particles in another 5 cu ft of air. The combined count on the two plates applies to 5 cu ft

of air. The electrostatic field of approximately 7000 volts between the electrodes in each unit is supplied by a rectifier circuit incorporated in the portable device. High resistances in the leads to the electrodes remove the hazards of dangerous electrical shocks. This device is known as the Luckiesh-Holladay-Taylor Electrostatic Bacterial Air-Sampler. Since the conical upper electrodes are also charged, they also collect bacteria. If identical bacteria-laden air enters both sampling units, it is reasonable to assume that the upper electrodes collect as many bacteria as are deposited on the petri dishes. This is taken into account in evaluating the results, since it is assumed that the sum of the catches on the two petri dishes represents the bacterial content of *one-half* the total air passing through the sampler. Careful tests indicate that the bacteria collected on the upper electrodes do not appreciably contaminate subsequent samplings. Consequently, it is generally unnecessary to sterilize or clean the sampler between samplings. Experimental evidence indicates a high efficiency

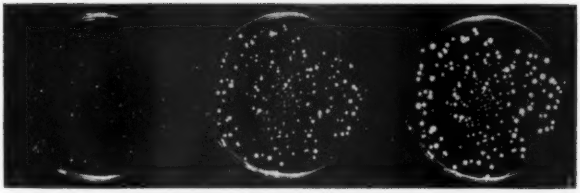


FIG. 4. ILLUSTRATING A PETRI DISH AFTER INCUBATING FOR 24, 48 AND 72 HR AT 37.5 C. ORGANISMS WERE COLLECTED FROM AIR IN AN OCCUPIED ROOM.

of collection. It is probable that naturally air-borne bacteria sometimes occur in *clumps*. It is improbable that such clumps are broken up in the process of collection; hence, only a single colony will develop upon incubation of such a clump.

GROWTH OF BACTERIAL COLONIES

Many types of nutrient media are commercially available; no single medium is satisfactory for all micro-organisms. For the collection of naturally air-borne bacteria, the authors have principally used Tryptose Blood Agar (Difco). After collection, the petri dishes are incubated for approximately 48 hr at 37.5 C, the most favorable temperature for colony growth. The rate of growth in a given case is illustrated by Fig. 4 from photographs of the same petri dish after incubation periods of 24, 48 and 72 hrs, respectively. The counts at 48 and 72 hr do not greatly differ, but the count at 24 hr is very much less. While the Tryptose Blood Agar is one of the best available media for growing most of the air-borne bacteria, Schneiter, Dunn and Caminita[8], of the *U. S. Public Health Service* have recommended Proteose Extract Agar as being superior to other media for the growth of several strains of streptococcic organisms which have been suggested as indices of air pollution.

SAMPLING PROCEDURE

Dust on the floor generally contains many bacteria. It is important to keep this fact in mind, since variations in activity of the occupants, or stirring of the air during the sampling period, may greatly influence the results and lead to erroneous conclusions. For example, in the first sampling, which the authors carried out in a motion-picture theater, the greatest concentration of bacteria was found just prior to the showing of the first picture, while the audience was entering the theater.

In determinations of bactericidal effectiveness of an installation of germicidal lamps, the authors have obtained the most reliable results by sampling during a period of minimum or nearly constant activity. The usual method is to have the lamps turned off for several hours before beginning the sampling, and then take samples for several 10-min periods during the hour before, and for at least an hour after, turning on the lamps. The petri dishes are then incubated for 48 hours before counting the colonies.

TABLE 1—A SUMMARY OF RESULTS OF SAMPLING AIR FOR BACTERIAL CONTENT IN VARIOUS LOCATIONS, ALL WITHOUT GERMICIDAL LAMPS.

LOCATION	COLONIES[a] PER CU FT OF AIR	
	Max	Avg
Large cafeteria during lunch period, six days in midwinter	83	44
Theater of medium size with small volume of air per person, Part of air cooled and recirculated. Summertime	74	30
Doctor's reception room, 5 to 15 patients	47	26
Dental operating room	41	32
Hospital A		
Surgical room, just vacated and washed	....	13
Delivery room, vacant	....	15
Infants' nursery	....	18
Cafeteria	....	25
Doctors' dressing room	....	420
Five public schools, summer	134	47
Orphanage, contagion ward, 8 patients	59	44
Children's home nursery for children of pre-school age	194	135
Dairy barn	....	2500
Poultry house	....	4000

[a]All these colonies were collected and grown on Tryptose Blood Agar (Difco) nutrient.

CONCENTRATION OF AIR-BORNE ORGANISMS IN VARIOUS PLACES

During a period of about 15 months the authors have sampled air in a large number of places, some with and others without germicidal installations. As a consequence, a great deal of information has been accumulated to show the concentrations encountered in various locations. Some of these data are summarized in Table 1.

Table 1 illustrates the fact that the concentration of air-borne bacteria varies greatly in various locations. It also varies quite appreciably at the same location on different days. Bacterial air-sampling is not an exact science, and unless the concentration is quite high it appears to vary appreciably over short periods.

The concentration of air-borne bacteria in a cafeteria during the lunch period is illustrated graphically in Fig. 5. Samplings over 10-min periods were made with the electrostatic sampler, and open dishes were exposed for sedimentation tests for periods of 30 min each. On the average over the whole sampling

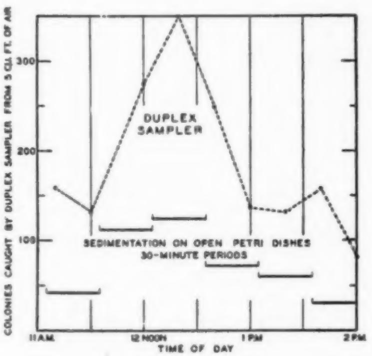


FIG. 5. SHOWING VARIATION IN CONCENTRATION OF AIR-BORNE ORGANISMS IN CAFETERIA BEFORE, DURING AND AFTER PERIOD OF HIGH OCCUPANCY. NUMBER OF ORGANISMS CAUGHT IN 10 MIN WITH DUPLEX ELECTROSTATIC AIR-SAMPLER IS COMPARED WITH NUMBER CAUGHT BY SEDIMENTATION ON OPEN PETRI DISHES IN 30-MIN PERIODS.

period, the sampler collected 7½ times as many bacteria per minute as were collected by the sedimentation plates. Obviously the electrostatic sampler yields data pertaining to the concentration of air-borne bacteria or the average number per cubic foot of air.

One of the most important possible places of infection by air-borne bacteria is the reception room of almost any doctor, where prospective patients wait for periods which may be one hour or longer. Samplings were made over a period of several hours in a reception room, with results as illustrated by the petri dishes in Fig. 6 and the data in Table 1. The sum of the colonies on the two plates in each column represents the micro-organisms collected from 5 cu ft of air. The dishes shown are representative of the minimum, average and maximum during the period of 2½ hr.

RESULTS OF GERMICIDAL INSTALLATIONS

With the assistance of other members of the laboratory staff, and using the procedure outlined, the authors have determined the effectiveness of germicidal lamps in a number of schools, hospitals, a dairy barn, a poultry house, etc. In some of the better installations, the following results were obtained: three schools, reductions of 45, 50 and 65 percent; dairy farm, cow barn, 55 percent; calf barn, 70 percent; poultry house, 75 percent. In several cases negative results were obtained, *i.e.*, the concentration was greater when the lamps were

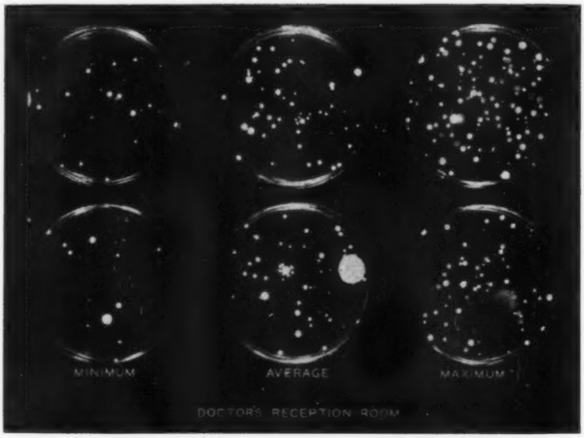


FIG. 6. PETRI DISHES, SHOWING COLONIES DEVELOPED FROM AIR-BORNE BACTERIA COLLECTED IN A DOCTOR'S RECEPTION ROOM OCCUPIED BY A NUMBER OF PATIENTS.

on than when they were off. Obviously this result was due to factors which overshadowed the lethal effect of the lamps. In two schools where the tests showed the lamps to be ineffective, the number of lamps installed was fewer than recommended[9]. In a nursery having a low ceiling, the lamps were installed only about 1 ft below the ceiling. This clearance is too small for effective irradiation.

Since excessive exposures to germicidal ($\lambda 2537$) energy produce erythema and conjunctivitis, (a temporary redness of the skin and irritation of the eyes, respectively), the lamps are usually installed in occupied rooms to irradiate the upper air of the enclosure above eye-level. Fortunately, very few paints used on walls and ceilings are efficient reflectors of this energy,[10] so that the energy reflected from the irradiated areas will usually be less than 10 percent of that incident upon them. If necessary, a paint with a low reflectance can be used.

The most efficient installation will be one which concentrates the maximum amount of energy in the upper air of the room, projecting it across the room so as to produce lethal dosages throughout the volume of air irradiated. If too

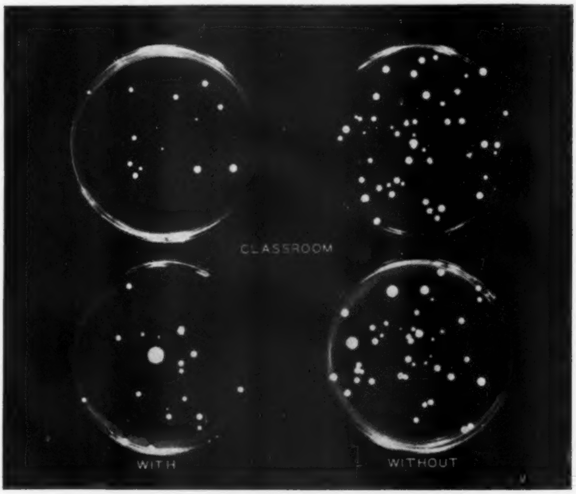


FIG. 7. RESULTS OF SAMPLING AIR IN SCHOOLROOM WITH AND WITHOUT GERMICIDAL LAMPS IRRADIATING THE UPPER STRATUM OF AIR. FOUR 30 WATT LAMPS IN A ROOM 20 FT x 40 FT WITH 12 FT CEILING.

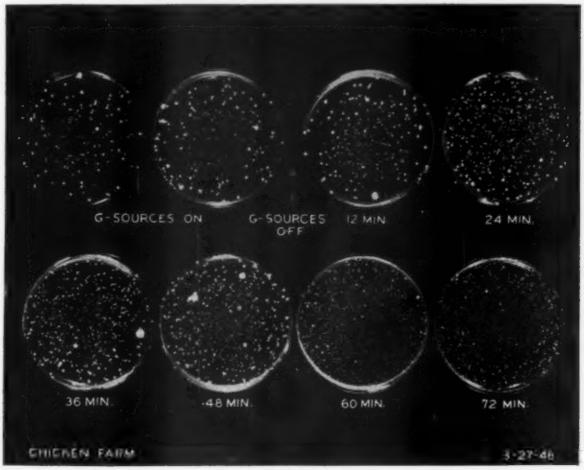


FIG. 8. PETRI DISHES EXPOSED TO SAME VOLUME OF AIR IN POULTRY HOUSE WITH AND WITHOUT GERMICIDAL LAMPS IRRADIATING MOST OF THE AIR. SAMPLINGS AFTER TURNING OFF LAMPS INDICATE INCREASING BACTERIAL CONTENT OF AIR.

much of the energy strikes the ceiling the installation will be inefficient, for too much of the energy will be absorbed without irradiating a large volume of air. Circulation of air in the room is essential, for the irradiated upper air must return to the lower occupied region in order to reduce the concentration of organisms in the breathed air. The maximum amount of energy which can be used in an occupied enclosure is limited by the permissible intensity of reflected energy reaching the eyes of persons in the room. Tentatively, the safe values appear to be 0.5 and 0.1 microwatt per square centimeter for 8-hr and 24-hr occupancy, respectively.

Air circulation greatly aids in moving the micro-organisms into the upper air where the germicidal energy can kill them. In winter the air currents pro-

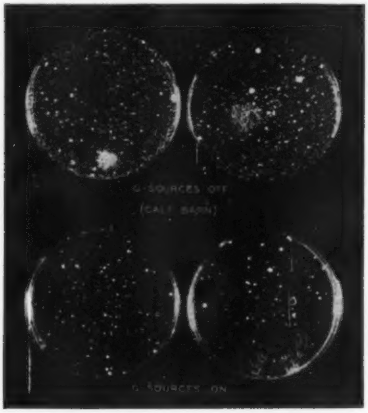


FIG. 9. AIR-BORNE ORGANISMS IN CALF BARN WITH AND WITHOUT GERMICIDAL LAMPS.

duced by the heating system aid in accomplishing this. Additional means of stirring up the air might be beneficial. Obviously, in many cases, germicidal lamps can be installed in air-ducts.

Some air-sampling tests were made at an underground poultry house having forced ventilation. Germicidal lamps had been installed along the corner between wall and ceiling at one side of the enclosure, but were baffled so that only about one-third of the floor area was irradiated. The rate of settling of dust on the floor was very high. The bacterial concentration, also was very high, too high for satisfactory sampling. As might have been expected, the lamps did very little good, since such a large part of the floor area, with high dust and bacterial contamination, was not directly irradiated.

If germicidal lamps of adequate wattage and number are properly installed, then a worthwhile reduction in air-borne organisms results. This is demonstrated by Fig. 7. Fig. 8 illustrates the rate of increase of organisms in a

poultry house after the germicidal lamps were turned off. The effectiveness of the lamps is quite evident. About an hour passed before the concentration of air-borne bacteria increased to an equilibrium value. Figs. 9 and 10 demonstrate quite strikingly some results achieved at a dairy farm. Since direct irradiation can be used, a greater reduction can be obtained than in schoolrooms, etc., where the irradiation must be confined to the upper part of the room.

Proper design of installations of germicidal lamps requires knowledge regarding the lethal effectiveness of the energy with various types of micro-organisms. At present, quantitative data along these lines are limited, but the authors and their colleagues are intensively studying this problem. For completeness, such a study should include many types of organisms which are commonly air-borne.

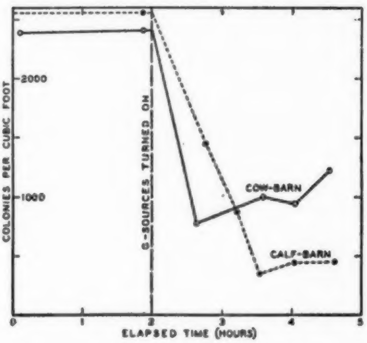


FIG. 10. GRAPHICAL ILLUSTRATION OF EFFECT OF GERMICIDAL LAMPS IN REDUCING BACTERIA IN A DAIRY.

This is a large undertaking and will require considerable time. Present engineering practice can undoubtedly be improved when more complete fundamental knowledge is available.

The authors are indebted to T. Knowles for valuable assistance in the researches leading up to the development of the air-samplers, and in the carrying out of many of the field tests. He has also done all the photographic work. They also gratefully acknowledge the aid of F. C. Kautzky who, for eight months, has ably assisted in the laboratory work and carried out the tests necessary for the identification of some specific colonies of micro-organisms.

REFERENCES

1. Air-Borne Infection as a Basis for a Theory of Contagion, by William F. Wells and Mildred W. Wells. (Aerobiology—Publication No. 17 of the *American Association for the Advancement of Science* 1942, 99.)

The Transmission of Certain Infections of Respiratory Origin, by Leon Buchbinder. (*Journal of the American Medical Association,* February 28, 1942, 718-730.)

2. Application of Germicidal, Erythemal and Infrared Energy, by Matthew Luckiesh. (D. Van Nostrand Co., New York, 1946.)

3. Studies of Air-Borne Virus Infections II. The Killing of Virus Aerosols by Ultraviolet Radiation, by D. G. Edward, Dora Lush and R. B. Bourdillon. (*Journal of Hygiene* 43, 1943, 11.)

4. A Comparative Study of Sampling Devices for Air-Borne Micro-Organisms, by H. G. duBuy, Alexander Hollaender and Mary D. Lackey. (Supplement No. 184 to the Public Health Reports, *U. S. Public Health Service*, 1945.)

5. Sampling Air for Bacterial Content, by Matthew Luckiesh, L. L. Holladay and A. H. Taylor. (*General Electric Review*, March 1946, 8-17.)

6. Catching Air-Borne Micro-Organisms, by Matthew Luckiesh and A. H. Taylor. (*The Magazine of Light*, June 1, 1946, 32-34.)

7. Sampling Devices for Air-Borne Bacteria, by Matthew Luckiesh, A. H. Taylor and L. L. Holladay. (*Journal of Bacteriology*, Vol. 52, No. 1, July 1946, 55.)

8. Studies in Connection with the Selection of a Satisfactory Culture Medium for Bacterial Air Sampling, by Roy Schneiter, John E. Dunn and Barbara H. Caminita. (Public Health Reports, *U. S. Public Health Service*, Vol. 60, No. 28, July 13, 1945, 789-806.)

9. Germicidal Fixtures, Basic Design and Use, by L. J. Buttolph and Howard Haynes. (*Lamp Department Bulletin LD-15*, General Electric Co.)

10. Transmittance and Reflectance of Germicidal (λ2537) Energy, by Matthew Luckiesh and A. H. Taylor. (*Journal of the Optical Society of America*, 36, April 1946, 227.)

DISCUSSION

DR. C.-E. A. WINSLOW, New Haven, Conn. (WRITTEN): I have been deeply interested in the subject of air-borne infection for many years. I devised one of the earliest bacteriological air-samplers, a very crude one, and the impinger method was evolved (for the determination of atmospheric dusts) by Leonard Greenburg, M.D., in my laboratory. I therefore welcome this important contribution to the subject by Messrs. Luckiesh and Taylor, and congratulate the Society on having it presented before us.

There are three major steps which must be taken if the problem of controlling air-borne disease is to be solved on sound engineering lines.

In the first place, we must have generally accepted standard methods for judging the efficiency of a given mode of disinfecting air. The remarkable progress of water bacteriology has been due to the acceptance of such standard methods which—while not perfect—have made possible intelligent comparison of different procedures of purification. We must agree on such standard methods for sanitary air analysis. Are we to measure pollution by the total number of colonies developing on a certain medium—and, if so, what medium? Or shall we use the acid-forming streptococci of the mouth as an index just as the colon bacillus of the intestine is used to measure pollution of water? I suspect that the results reported here today (with purification effectiveness ranging from nothing to a maximum of 75 percent) might have been much more favorable if streptococci, instead of total colonies, had been used as measuring sticks. Shall we collect our samples by impingement alone or by an electrostatic process? Shall we use the Wells centrifuge, the Hollaender aeroscope, or the Luckiesh electrostatic device? If different procedures are employed, results will differ widely and will not be in any sense comparable. It is of little consequence what proportion of the microbes in the air is collected, for no device collects them all. What we need is a method which gives us the ones which indicate most sharply the difference between good air and bad air; and above all what we need is agreement, for the time being, on one single standard procedure which will yield comparable results. The *U. S. Public Health Service* or the *American Public Health Association* should give us a standard procedure.

When we have agreed upon a generally-accepted measure of air purity, the next step is to compare the effectiveness of various procedures for air purification, and their

costs. There are at least three methods of reducing the bacterial content of air which have their ardent advocates. They are the reduction of organisms contributed to the air by floors, bedding, and other natural objects, by treatment of these objects with dust-holding oily substances; the disinfection of the air itself by the use of aerosols; and the use of ultra-violet light. Which of these three procedures accomplishes maximum results at minimum cost, no one can say; yet that is what the engineer must know, in order to deal intelligently with the subject. He must know it, too, in relation to a particular problem. Oil treatment may be the method of choice for barracks, aerosols for a chicken house, ultra-violet for a school room.

Finally, it must be determined by the epidemiologist just how great the need is for air disinfection in a given space. All experts, I think, are agreed that some process of this kind is essential in the surgical operating room and in the contagious ward of a hospital. Beyond that, we cannot at present go with certainty. Experiments in army barracks during the war yielded conflicting results. Elaborate experiments are under way, in New York State and elsewhere, as to the actual effect of air disinfection in schools on the prevalence of respiratory disease; but these experiments will take several years to yield reliable results. At present, the most competent experts are not agreed that routine disinfection of the air of school rooms and auditoria is necessary or desirable. This subject is still in the experimental state; and the engineer should await the considered verdict of the experts in epidemiology before adopting any process as a general procedure.

W. L. FLEISHER, New York, N. Y. (WRITTEN): Over a period of years the Society has had papers presented on the effectiveness of radiant energy developed by lamps of definite wave length as a means of destroying air-borne bacteria or viruses. This year we again have a paper exploiting this theory.

I am always impressed by the ease with which half facts or unresolved results are correlated, with profound carelessness to establish conclusions which nothing in the empirical results warrants.

To me there are two interesting and salient observations that could be culled from this paper. One of them, of course, has been pertinent to every paper on this particular subject, and that is the fact that only a certain definite wavelength is lethal to bacteria and a virus. If we knew why this particular wavelength, which has an energy factor of a definite amount, was the maximum lethal agency, we might approach the question of our own energy equilibrium with a little more certainty. Of course, my objection to the vast propaganda of the electrical companies advocating the use of sterile lamps with wavelengths of 2600 Angstroms—to be exact, 2537—is due to the fact that this method has never really been practicable. Years ago it was introduced in bakeries for the killing of mold spores (which actually could be eliminated by proper filtering). However, the electrical wave is lethal only if it can definitely strike the bacteria or virus. So long as these micro-organisms are concealed in any way from the direct rays, they continue their immunity.

The authors of this paper advocate stirring the dust that settles on the ground in order that it may get into more intimate contact with the rays from these lamps, which necessarily must be above the sight line of the occupants of the building owing to the fact that exposure to germicidal energy produces erythema and conjunctivitis. The authors claim that air circulation aids in moving the micro-organisms into the upper air, where germicidal energy can affect the micro-organisms. It would be my feeling that, in developing this air movement, the germs or bacteria would be absorbed by the individuals in the enclosure and create more mischief than if they were allowed to remain quiescent.

Under the very best conditions, the radiation from the lamps can affect or destroy only a certain number of these micro-organisms, as indicated by the illustrations shown in the paper.

According to the authors, this entire subject is still in a very elementary and inconclusive state. If I may orient this method of air sterilization to the triethylene glycol or propylene glycol method investigated by Robinson of Chicago and the various members of the group at Northwestern, not to mention Dr. Stokes and Dr. Harris and his group at Philadelphia, I might say that, seemingly, saturation of the air with a vapor, such as the glycol vapor, being unaffected by exposure and completely occupying the space so far as any molecules could occupy a space, should have better chance

of ultimate sterilization than the lamp method, which is so dependent on accuracy of aiming.

However, in conclusion, I believe that a study of the kinetic energy of the particular wavelength which has been proved lethal may lead to some elementary information which could be utilized to the advantage of the human race.

C. H. WHITE, New Bedford, Mass.: I am interested in the warm air heating industry, particularly forced warm air heating. In the course of this heating work, I have been asked, several times, to heat small apartment houses with forced warm air. I have shied away from that for the reason that I have been afraid that if I recirculated air from each apartment, and if, in one of those apartments there happened to be some contagious disease, I might spread that disease throughout the rest of the apartments. I would be interested to know from the speaker whether that is possible and, if so, whether there is anything that could be used in the filtering system of that air which returns it to the apartment house that would eliminate the possibility of contagion in the other apartments.

M. H. KLIEFOTH, Madison, Wis.: I would like to know what has become of the research work which was done during the war by the Army Surgeon General, regarding cross infectional diseases, and the tests which were made, with mechanical air filtration, in regard to the reduction of bacteria by mechanical filtration.

DR. WINSLOW: As to those results, some of them have been published, and others will be published in the *American Journal of Public Health.* The results to which you refer are in issues for the first three months of 1947.

R. L. KUEHNER,* York, Pa.: We have had occasion in the past few months to go into this business of electrical charge on particles suspended in the atmosphere, and have found not only that there are positively and negatively charged ones but that in addition, there are those which have no charge or are neutral.

In Mr. Taylor's discussion, I was wondering if, in any way not explained in the paper, he has taken into account these neutral particles, as we expect to get adsorption of bacteria on those, as well as on the positively and negatively charged particles. The collections, which he discussed, referred only to positively and negatively charged particles.

As Dr. Winslow has stated, the big value of any method of collecting bacteria is in the comparability. We have found that the neutral ions or the air particles with no charge, vary in relative and total concentration and do so within short periods of time. We recognize that variation in the distribution of positively and negatively charged particles has no effect since, theoretically, all of these are captured. However, if bacteria are adsorbed upon neutral particles which are not captured, and these particles are markedly variable, there goes the comparability.

My other question is, have you, Mr. Taylor, any figures comparing your apparatus with some of the more commonly used techniques evaluating air-borne bacteria?

S. KONZO, Urbana, Ill.: In view of the fact that the electrostatic precipitator, as used in the sampler, showed evidence of reducing the bacteria count, why could not the electrostatic precipitator itself be used as a means of reducing bacteria?

AUTHOR'S CLOSURE: We want to thank Dr. Winslow for his very helpful and interesting discussion of our paper. We have already considered most of the points he raised, but lack of time prevented their discussion in my presentation of the paper.

Unfortunately, there is no simple index of air contamination by respiratory organisms. Dr. Buchbinder and some others have used alpha haemolytic streptococci as a measure of air contamination, since they are easily identified on a suitable medium. However, there is no universal agreement on this organism, as was evident in a panel discussion at the recent meeting of the *Society of American Bacteriologists* in Philadelphia. There is need for further study of sampling methods and culture techniques for the purpose of developing test methods which will achieve a differentiation between respiratory and non-respiratory organisms.

In any closed room with several occupants, such as a schoolroom, there are certain to be some respiratory organisms in the air, but they may compose only a small percentage of the total number. Consequently, as Dr. Winslow pointed out, a measured

*Bacteriologist, Research Department, York Corp.

reduction of the total organisms by use of ultraviolet irradiation of the air may actually indicate a much greater reduction of streptococci and other respiratory organisms.

In some recent work, completed after this paper was presented (see *Journal Franklin Institute* 24, 1947, 267), we have approached the problem much more directly. By atomizing saliva into the air in a closed room for a while then treating the air with germicidal energy, a high initial concentration of respiratory organisms and a very high rate of killing were measured. In other laboratory studies it was found that the ultraviolet dosage required to kill the saprophytic organisms was 500 to 1000 times as great as that required to kill respiratory organisms in atomized saliva.

In our work so far we have not attempted a comprehensive study of results obtained by different types of air-samplers. Our objective was the development of a simple portable air-sampler having good efficiency, and we believe that our electrostatic air-sampler fulfills these requirements. It is very improbable that it breaks up clumps of bacteria such as may occur on dust particles, hence it would undoubtedly show a lower count than would the Moulton atomizer-bubbler sampler which does break them up.

We have run numerous parallel tests between the radial jet air-sampler, Fig. 2, and the electrostatic sampler, Fig. 3. The average of many tests indicates that their efficiencies are the same within less than 5 percent. In another test an electrostatic sampler was connected in series with the radial jet sampler, to collect any organisms which passed through it. The second sampler caught less than 5 percent as many as were caught by the radial-jet sampler (Fig. 2). Fig. 5 shows that in the cafeteria tests the electrostatic air-sampler collected, in 10 minutes, approximately 3 to 4 times as many organisms as were collected on sedimentation plates in *30* min.

Some comparisons were made with the Hollaender-type funnel aeroscope. The highest collection by the latter was only about ½ that of our sampler, and sometimes it was far less than that.

As Dr. Winslow has pointed out, no sampling method collects all the bacteria. Furthermore, the efficiency of any type of sampler probably depends to some extent on the way in which the micro-organisms are suspended in the air, i.e., whether on dust particles or in minute droplet nuclei. While we cannot assign a definite efficiency value, such tests as we have made indicate that it is very high. In the case of organisms introduced into the air by nebulizing saliva, it was found that the use of the electrostatic field multiplied the efficiency of collection thirty times.

Tests of air-disinfection by germicidal lamps in army and navy camps during the war and since have shown a consistent reduction of the order of 25 percent in respiratory diseases, when non-irradiated barracks were used as controls. From an economic standpoint alone, a definite reduction of even as little as 5 to 10 percent in absences of employees due to respiratory diseases would in many offices more than pay for the cost of installation and maintenance of germicidal lamps.

Mr. Fleisher intimates that the conclusions arrived at in this paper and in the summary are not warranted by the facts presented. For example, he objects to the statement in the summary *Bacteria and viruses regardless of their physical nature can be destroyed by suitable exposure to ultraviolet energy.* The authors have studied the effect of germicidal energy on many specific types of micro-organisms as well as the mixed types found in very dusty air. We have not found any bacteria which could not be inactivated by germicidal ultraviolet, although the required dosages vary over a wide range. Our work has not included studies of viruses, but published and unpublished work indicate that the same statement holds true for them. Consequently, the authors wish to state that they emphatically agree with the above quoted statement. Obviously, if the organisms are in a liquid medium opaque to the germicidal energy they cannot be killed by it.

Two other points raised by Mr. Fleisher are not justified by anything presented in the paper, as a careful reading of it will show. *First* is his statement that *only a certain definite wavelength is lethal to bacteria and a virus.* Energy throughout a wider spectral range than that shown in Fig. 1 is lethal to these organisms. However, just as in the case of vision, a narrow wavelength range is more efficient than any other. For the destruction of micro-organisms this range centers very nearly at the wavelength of maximum output of the germicidal lamps, 2537. Sunlight and skylight are bactericidal agents, but they contain no energy of wavelengths shorter than approximately λ2950. Experiments in our laboratory with E. coli have shown that equal amounts of killing of this organism require over five thousand times as much

energy with mid-summer sunlight as is required with energy from germicidal lamps.

Secondly, we do *not* advocate stirring up the dust to bring it into the irradiated zone. Dust in an occupied interior is unavoidably stirred up by air drafts and movements of the occupants, and it is important to kill any pathogenic organisms carried by the dust. This will be accomplished in the irradiation zone.

In any method of air disinfection it is unlikely that all air-borne organisms will be destroyed, but this is no more necessary than is the complete sterilization of milk or drinking water.

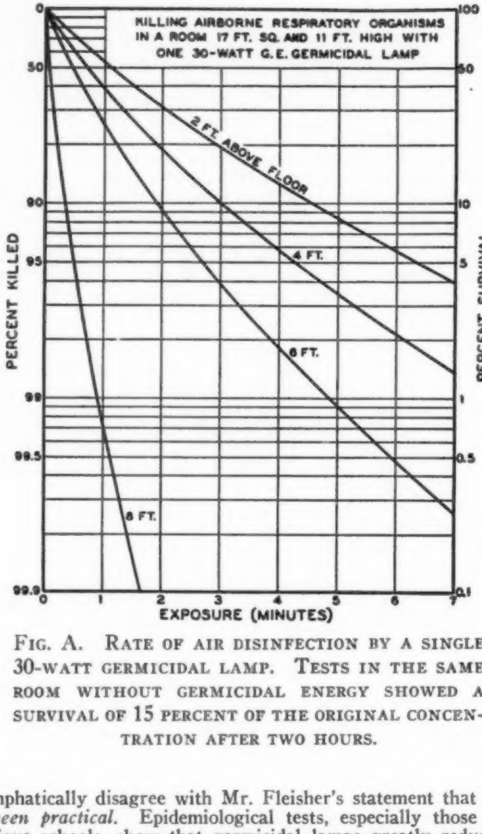


FIG. A. RATE OF AIR DISINFECTION BY A SINGLE 30-WATT GERMICIDAL LAMP. TESTS IN THE SAME ROOM WITHOUT GERMICIDAL ENERGY SHOWED A SURVIVAL OF 15 PERCENT OF THE ORIGINAL CONCENTRATION AFTER TWO HOURS.

We most emphatically disagree with Mr. Fleisher's statement that *this method has never really been practical.* Epidemiological tests, especially those reported on by Wells for various schools, show that germicidal lamps greatly reduce the spread of epidemic diseases. While it is true that the organisms must receive direct irradiation in order to be killed, we have found that the required dosage for organisms of respiratory origin is very small. Certainly our results show the method to be decidedly effective and practicable.

A proper installation of germicidal lamps in any occupied interior irradiates directly only the upper air in the room above eye-level. In a closed room, with only one occupant, we recently measured air-currents in various parts of the room and found that the currents were predominantly upwards, 25 to 50 fpm, in most areas of the room

excepting the corners, where the air was moving downwards. In an occupied room the warm breath and heated bodies of the occupants add considerably to the tendency of the air to move into the upper, irradiated zone, thus increasing the air circulation. Actually the decrease in the concentration of air-borne organisms outside the irradiation zone is adequate proof of rapid mixing of the air.

Very little germicidal energy is reflected from the ceiling and walls into the lower part of the room. However, even the low intensity at eye level specified by the

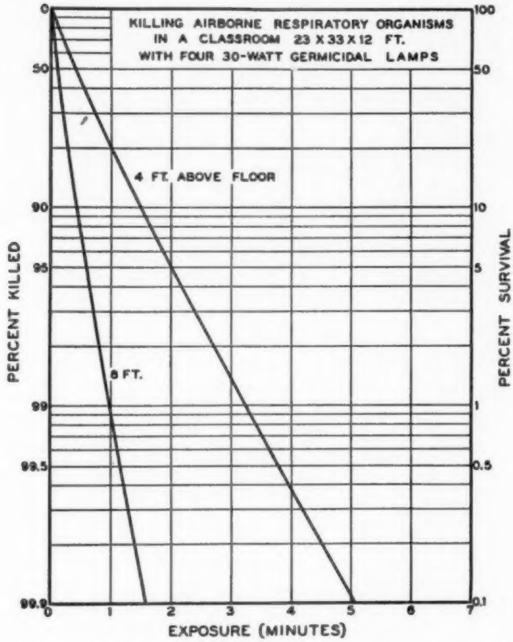


FIG. B. TESTS SIMILAR TO THOSE OF FIG. 11, CARRIED OUT IN A SCHOOL CLASSROOM WITH FOUR 30-WATT GERMICIDAL LAMPS.

American Medical Association, 0.5 microwatt per sq cm for 7 hr daily use, will kill 80 percent of the respiratory organisms in two minutes.

Our tests in an experimental room and a schoolroom amply verify the above statements. Fig. A shows results achieved at four levels in the center of a room 17 ft square and 11 ft high, using only one 30-watt germicidal lamp in a louvered reflector 7 ft above the floor. Saliva was nebulized into the room for about ½ hr and then stopped. The air was then sampled and the lamp started. The rate of reduction of respiratory organisms was very high at all levels, especially at 8 ft. A similar test in a schoolroom, Fig. B, gave comparable results, but in this case the nebulization of saliva was continued while the lamps were burning, putting into the air approximately a million micro-organisms per minute.

While we have clearly shown that the contamination of air by respiratory organisms can be reduced to a very low level by germicidal lamps, it is very difficult to measure

the effect of this reduction upon the health of occupants of the room. Complete control of occupants is obviously rarely possible.

While we have had no first-hand experience with the glycols, it seems improbable that they could achieve any higher degree of air disinfection than is indicated by Figs. A and B for germicidal lamps.

Mr. White asks regarding the disinfection of recirculated air in small apartment houses. It is entirely practicable to do this by installing germicidal lamps in the air ducts. Complete engineering data for such installations are already available.

In reply to Mr. Kuehner's question as to the collection of uncharged particles, we think that such particles would be non-selectively collected in both dishes. It is a well established fact that if a neutral particle is brought near a charged surface, the nearer side develops a charge of opposite sign while an equal charge of the same polarity as the surface is developed on the opposite side. This would result in the dust particle or other object, whatever its nature, being collected by the electrode to which it is nearest. The treatment is purely theoretical since we have no experimental proof that collection in the sampler takes place in that way.

Mr. Konzo asks why the electrostatic precipitator itself could not be used as a means of reducing bacteria. It probably could, but we doubt whether its effectiveness would be as great as that of a well-designed installation of germicidal lamps.

1314

No. 1314

SEMI-ANNUAL MEETING, 1947

Coronado, Calif.

WITH an attendance of 240, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Semi-Annual Meeting, 1947, at the Hotel del Coronado, Coronado, Calif., June 2-4, provided an opportunity for reviewing the current status of ASHVE research, with a spirited discussion of comfort standards and physiological studies and comments on panel heating investigations. After the technical sessions, the Committee on Arrangements of the Southern California Chapter provided an enjoyable program of social and sports events.

FIRST SESSION—MONDAY, JUNE 2, 10:00 A.M.

The meeting was opened by Pres. B. M. Woods, Berkeley, Calif. Art Theobald, president of the Southern California Chapter, gave a brief message of welcome to which Pres. Woods responded.

R. A. Lowe, chairman of the Committee on Arrangements, was introduced and outlined some of the special program features.

The first technical session was conducted in the ballroom of the Hotel del Coronado, Coronado, Calif., June 2, by A. E. Stacey, Jr., second vice president, Syracuse, N. Y., who introduced Cyril Tasker, director of research, Cleveland, as the first speaker. His subject, The Challenge of the A.S.H.V.E. Research Program, gave the early history of the Laboratory, reviewed the status of the current program and announced the availability of a bibliography of published data on panel heating.

H. B. Nottage, Cleveland, Ohio, presented his paper, illustrated by sides, on The Thermal Properties of Building Materials Used in Heat Flow Calculations (see Chapter 1315).

The paper, Forced Convection Heat Transfer from Flat Surfaces, by G. V. Parmelee, and R. G. Huebscher (see Chapter 1316) was also presented by Mr. Nottage.

Vice President Stacey then called upon Prof. F. W. Hutchinson, Lafayette, Ind., who presented his paper by title on the Influence of Gaseous Radiation in Panel Heating (see Chapter 1317).

The technical session was adjourned at 12:00 noon.

SECOND SESSION—TUESDAY, JUNE 3, 9:30 A.M.

Pres. B. M. Woods called the meeting to order at 9:30 a. m. and announced that the session would be devoted to papers relating to human comfort and physiological reaction. Nathaniel Glickman, Chicago, Ill., presented the first paper, Physiological Adjustments of Human Beings to Sudden Change in Environment, by Mr. Glickman, Tohru Inouye, Stanley E. Telser, M.D., Robert W. Keeton, M.D., Ford K. Hick, M.D., and Maurice K. Fahnestock (see Chapter 1320).

The second paper, Conditions for Comfort, by C. S. Leopold, Philadelphia, Pa., (see Chapter 1318) was presented by the author.

Dr. Woods introduced Prof. R. C. Jordan, Minneapolis, Minn., who presented the paper, Comfort Reactions of 275 Workers During Occupancy of Air Conditioned Offices, by F. B. Rowley, R. C. Jordan, and W. E. Snyder (see Chapter 1321).

The final paper of the session, A Method for Improving the Effective Temperature Index, by C. P. Yaglou, Boston, Mass., was presented by John Everetts, Jr., (see Chapter 1319).

In presenting the paper, Mr. Everetts pointed out that although recent studies in body heat regulation have not developed a more practical thermal index than effective temperature (ET) much fundamental information has been contributed that can be used to correct the ET index for humidity and radiation.

The second technical session was adjourned at 12:00 noon.

THIRD SESSION—WEDNESDAY, JUNE 4, 9:30 A.M.

The technical session was called to order at 9:30 a.m. by President Woods. The first paper, Experimental Studies on Panel Heating Tube Spacing, by B. F. Raber and F. W. Hutchinson, was presented in brief by Professor Hutchinson (see Chapter 1322).

President Woods then requested L. P. Saunders, Lockport, N. Y., chairman of the ASHVE Committee on Research, to take charge of the meeting. Mr. Saunders presented Prof. B. H. Jennings, Evanston, Ill., who gave the paper Triethylene Glycol Vapor Distribution for Air Sterilization, by Edward Bigg, M.D., B. H. Jennings, and F. C. W. Olson (see Chapter 1324).

The next paper, Determination of Outside Design Temperatures, by W. L. Holladay, Los Angeles, Calif., (published in the July 1947 A.S.H.V.E. JOURNAL SECTION, *Heating, Piping & Air Conditioning*) was presented by the author.

The author suggested a method of using present available Weather Bureau temperature records for establishing outside design temperatures for summer and winter. He included nomographs for determining (1) winter design temperatures from the average minimum and lowest temperatures, and (2) summer design temperatures from average maximum temperature and highest temperature.

The final paper of the session, The Effect of Floor Slab on Building Structure Temperatures and Heat Flow, by Prof. C. F. Kayan, New York, N. Y. (see Chapter 1323), was presented by the author.

President Woods returned to the chair and called for a report of the Resolutions Committee, which was presented as follows:

Resolutions

WHEREAS, the 1947 Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in Coronado, Calif., takes an enviable place in the annals of our Society, because of the job so well performed by the Southern California Chapter, its Committee on Arrangements, and many of its members, and

WHEREAS, many other individuals and organizations have also contributed much to this very successful meeting, therefore,

BE IT RESOLVED, That theAMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS adopt an expression of appreciation of the work they have done, and that copies be sent to the following:

* * *

To Pres. Art Theobald and the Officers of the Southern California Chapter,

To R. A. Lowe, chairman of the Committee on Arrangements, to its members and their ladies,

To the authors of the technical papers,

To Leo Hungerford, toastmaster at the Banquet,

To Dr. Stafford Warren for his address at the Banquet,

To the Hotel del Coronado, its management and staff for their gracious hospitality and service,

* * *

To the members of the Barber Shop Quartette, Messrs. Allen, Miller, Moffett and Snyder for their outstanding performance, both musically and as chefs de cuisine at the beach party,

To the newspapers and trade publications for their coverage of the meeting,

To the San Diego Convention Bureau for its cooperation in preparing for the meeting,

To the Chapters and members who continue to contribute toward the goal of retiring the Research Laboratory Mortgage.

Respectfully submitted,

RESOLUTIONS COMMITTEE

C. Rollins Gardner, *Chairman*
Melville G. Kershaw
Leon T. Mart

The technical session adjourned at 12:00 noon.

On Tuesday evening the Semi-Annual banquet was held and the speaker, Dr. Stafford Warren, dean of the Medical School, University of California in Los Angeles, told some of the little known details of atomic bomb effects. At the banquet Leo Hungerford was toastmaster. President Woods presented the Memory Book to Past President A. J. Offner. L. P. Saunders presented the Research Cup to M. F. Blankin, and Mr. Blankin presented the Eichberg Cup to Art Theobald, president of the Southern California Chapter.

PROGRAM—SEMI-ANNUAL MEETING

American Society of Heating and Ventilating Engineers

Hotel del Coronado, Coronado, Calif.

June 2-4, 1947

SUNDAY, JUNE 1

10:00 a.m. Registration—Lobby, Hotel del Coronado

2:00 p.m. Council Meeting—Reading Room

7:00 p.m. Nominating Committee

MONDAY, JUNE 2

9:00 a.m. Registration—Lobby

10:00 a.m. RESEARCH SESSION—Ball Room—2nd Vice President A. E. Stacey, Jr., Presiding

GREETINGS—By Art Theobald, Pres. So. California Chapter

RESPONSE by Dr. Baldwin M. Woods, President ASHVE

Challenge of the A.S.H.V.E. Research Program, by Cyril Tasker, Director of Research

The Thermal Properties of Building Materials Used in Heat Flow, Calculations, by H. B. Nottage

Forced Convection Heat Transfer from Flat Surfaces, by G. V. Parmelee, and R. G. Huebscher

Influence of Gaseous Radiation in a Panel Heating, by F. W. Hutchinson

10:30 a.m. Children's Sports—Playgrounds

12:30 p.m. Luncheon

12:30 p.m. Deep-Sea Fishing Trip

2:00 p.m. Roads to Romance—Old Town—Ramona's Wedding Place—Balboa Park
Golf Tournament—Research Cup and Eichberg Memorial Cup, San Diego Golf Club

Beach Party for Children

6:30 p.m. Dinner—Main Dining Room

8:30 p.m. Chapter Delegates' Meeting—Gold Room

Movies

Surprise Party

TUESDAY, JUNE 3

9:00 a.m. Registration—Lobby

9:30 a.m. TECHNICAL SESSION—Ball Room—Pres. Baldwin M. Woods, Presiding

Conditions for Comfort, by C. S. Leopold

A Method for Improving the Effective Temperature Index, by C. P. Yaglou

Physiological Adjustments of Human Beings to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, Stanley E. Telser, M.D., Robert W. Keeton, M.D., Ford K. Hick, M.D., and Maurice K. Fahnestock.

Comfort Reactions of 275 Workers During Occupancy of Air Conditioned Offices, by F. B. Rowley, R. C. Jordan and W. E. Snyder.

10:30 a.m. Ladies' Card Party

12:30 p.m. Luncheon

2:00 p.m. Harbor Cruise of San Diego Bay, Point Loma and Navy Docks.

2:00 p.m. High Altitude Laboratory Demonstration—Consolidated Aircraft Co.

5:30 p.m. Social Hour—Swimming Pool Terrace

7:00 p.m. Banquet—Main Dining Room

Toastmaster: Leo Hungerford

Presentation of Memory Book to Alfred J. Offner by President Baldwin M. Woods

Speaker: Dr. Stafford Warren, Dean, Medical School, University of California, Los Angeles.

Subject: The Significance of the Bikini Tests to Engineering

Dancing in Circus Room

WEDNESDAY, JUNE 4

9:30 a.m. TECHNICAL SESSION—Ball Room—Pres. Baldwin M. Woods Presiding

Experimental Studies on Panel Heating Tube Spacing, by B. F. Raber and F. W. Hutchinson

Effect of Floor Slab on Building Structure Temperatures and Heat Flow, by C. F. Kayan

Triethylene Glycol Vapor Distribution for Air Sterilization, by Edward Bigg, M.D., B. H. Jennings, and F. C. W. Olson

Determination of Outside Design Temperatures—A Suggested Approach, by W. L. Holladay

12:30 p.m. Luncheon

2:00 p.m. Trip to Old Mexico, the Land Of Manana.

COMMITTEE ON ARRANGEMENTS

R. A. LOWE, *General Chairman*

J. L. BLAKE
R. S. FARR
LEO HUNGERFORD
MARON KENNEDY

Finance..............W. N. PAULEY, *Chairman*

Reception and Publicity
E. A. GUENTHER, *Chairman*

Sports Events....BOB JOHNSON, *Chairman*

Hotels and Reservations
A. G. OREAR, *Chairman*

Outside Entertainment
J. S. EARHART, *Chairman*

Ladies............A. B. BANOWSKY, *Chairman*

Transportation (Local)
C. W. POLLOCK, *Chairman*

Special Events....VERN MILLER, *Chairman*

Inspection Trips
MARON KENNEDY, *Chairman*

Industry Committee
L. B. DAVENPORT, *Chairman*

No. 1315

THERMAL PROPERTIES OF BUILDING MATERIALS USED IN HEAT FLOW CALCULATIONS

By H. B. NOTTAGE,† CLEVELAND, OHIO

This paper†† is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio

I. INTRODUCTION

AS a part of the A.S.H.V.E. research program of the last few years, analytical studies [1, 2] which were made on periodic heat flow through walls and roofs have emphasized the need for a practical evaluation of available data on the thermo-physical properties of building materials.

A survey has therefore been made of information readily available in the literature or obtainable from other reliable sources. The literature data selected have been those which, as far as available, apply to the conditions of heating and air conditioning applications. The data collected and the results of the studies on the thermo-physical properties of building materials have been summarized for ready reference in a series of tables and charts which are described in the sections of this Bulletin.

The thermo-physical properties needed for each material entering into a periodic heat-flow problem are:

1. Thermal conductivity.
2. Density.
3. Unit heat capacity (or specific heat).*

The magnitudes of these properties for any one material will be dependent upon the temperature and the moisture content.

Where a given type of material is further subject to variations in density, the thermal conductivity will also vary with the density. Unit heat capacity, on the other hand, is essentially only a characteristic of the kinds and amounts of the molecular species involved.

The building materials considered have been grouped for classification as:

1. Wood.
2. Building and insulating boards and insulating materials.
3. Masonry materials.
4. Concrete, plaster, and mortar.
5. Glass.
6. Roofing materials.

---

† Research Associate, A.S.H.V.E. Research Laboratory. Member of A.S.H.V.E.
†† Published as A.S.H.V.E. Research Bulletin, Vol. 53, No. 2.
[1] Exponent numerals refer to References.
* Specific heat and unit heat capacity are numerically equal; but specific heat is a dimensionless ratio while unit heat capacity has dimensions. The latter is therefore used in the calculations.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Coronado, Calif., June 1947.

The data have been summarized in a ready-reference table for each classification. A set of curves has also been included for the first group, *i.e.*, wood, to give complete data on the properties for different species, moisture contents, and temperatures.

To aid in the practical application of the data, there have been included in each table calculated magnitudes of:

1. The product (thermal conductivity) × (density) × (unit heat capacity), which is needed as a factor in periodic heat-flow design charts.
2. The thermal resistance per inch of material thickness.
3. The thermal diffusivity.

Discussions and comments on the available information for each classification of material are given in the corresponding section of the report.

II. Conclusions

The conclusions drawn as a result of this study are as follows:

1. The available data on thermal conductivity, density, and unit heat capacity are considered to be generally adequate for most load-estimating purposes for the following classes of materials:

   Wood.
   Glass.
   Building and insulating boards.
   Insulating materials.

2. The available data on thermal conductivity, density, and unit heat capacity seem to be sufficient for rough load-estimating calculations but not necessarily extensive and accurate enough for all types of refined analyses for the following classes of materials:

   Masonry.
   Concrete, plaster, and mortar.
   Roofing materials.

3. The lack of an extensive amount of accurate data for building materials in general, and particularly for those noted in Item 2, is largely due to the fact that since these materials as used in practice are subject to large variations in composition, state of aggregation, moisture content, and temperature, more detailed data covering the many variable conditions did not seem to have economic value.

III. Recommendations

1. The Technical Advisory Committee on Cooling Load recommends that the data presented in this Bulletin be used as the basis for practical design-estimate calculations to meet current needs in treating periodic heat flow through walls and roofs.

2. The committee believes that any experimental program undertaken to determine the thermo-physical properties of building materials should be planned on a comprehensive basis covering a wide array of test samples and range of test conditions. Such a program does not, however, appear to be necessary at the present time.

3. Should it later prove desirable to make experimental studies on periodic heat flow, the committee suggests that the properties of the specific materials employed be determined in order to supplement the data given in this Bulletin.

IV. COMMENTS ON DATA REQUIREMENTS FOR APPLICATION PROBLEMS

The first general requirement of an air conditioning load estimate is the exercise of good judgment. This applies with equal importance to the selection of design-load conditions and to the choice of the thermo-physical properties of the construction materials needed in the load calculations.

Building materials are either natural products, mixtures of various natural and prepared substances, or specially manufactured materials. The natural constituents, and those prepared from natural sources in particular, are subject to variations in their physical and chemical nature according to their source and method of preparation. Manufactured materials belonging to the same nominal trade classification also may have somewhat different physical and chemical natures. Furthermore, gradations in the compactness of a given material will produce differences in thermo-physical properties.

Moisture content and temperature also influence thermo-physical properties in various ways; in practice, these two variables are subject to uncontrolled weather disturbances. An unwieldly, impractical, and extensive store of data would be required if each property of each material were to be specified with high accuracy and all influences acting thereon taken into consideration. Such a store of data would, moreover, be without practical justification for air conditioning load estimates.

While it is proper to recognize that considerable variations may exist in the thermo-physical properties of any nominally specified material, the only essential for practical load estimates is a reasonable representative value for each property involved. The present compilation has been approached from a practical attitude. (It may be well to comment, however, that in many instances the selection of a *reasonable, representative value* has been reduced to the acceptance of the *only value* available.)

A word is in order regarding the relative requirements for data on each of the three properties † which are of importance to load-estimate problems, namely

Thermal conductivity.
Density.
Unit heat capacity.

The thermal conductivity is the most important single property; it enters into both the periodic and the steady-state phases of heat flow. Data on thermal conductivity should therefore be the best obtainable.

The density and unit heat capacity enter only into the periodic heat-transfer calculations. Since the periodic component of a load estimate is only a part of a more complete problem, it need not be treated with highly refined accuracy. Accordingly, while data on these two properties certainly should be the best

† The three properties are frequently reduced to two, the thermal conductivity and a derived property called the *thermal diffusivity*, $a = k/\rho c_p$. In this Bulletin it was considered preferable to place primary emphasis on the three individual properties noted, and then to consider as derived properties the two quantities:

$$(k\rho c_p) = (\text{thermal conductivity}) \times (\text{density}) \times (\text{unit heat capacity}), \text{ and}$$
$$a = (k/\rho c_p) = \text{thermal diffusivity}.$$

The product $(k\rho c_p)$ is employed in the Mackey and Wright[1] design charts, and the thermal diffusivity is widely used in time-variable heat-transfer analyses.

available, it may be permissible to employ approximate or estimated values when nothing else is obtainable.

V. NATURE OF THE DATA TABULATED

The data given in this Bulletin have limitations which must be recognized. They have been obtained either from readily-available references or collected from organizations in a position to supply pertinent information. They have

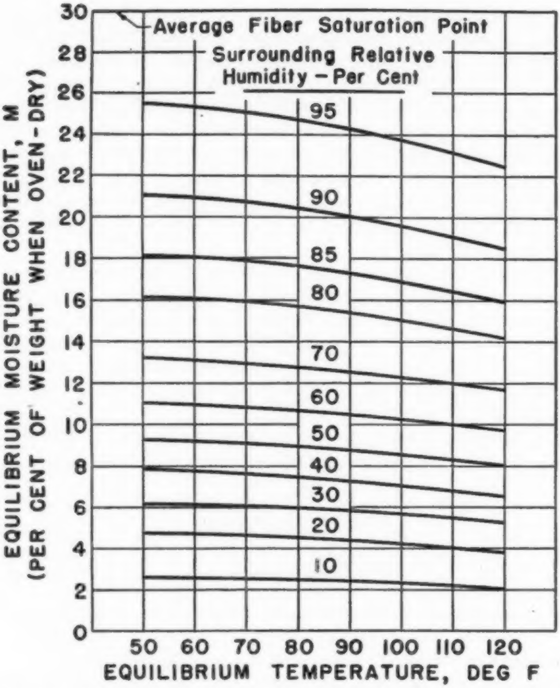


FIG. 1. EQUILIBRIUM MOISTURE CONTENT OF WOOD AT ATMOSPHERIC PRESSURE

not been traced back for an exhaustive analysis of the method by which they were established. This would have been time-consuming, and the data available proved to be so limited that an exacting judgment between competitive values was not required. Citation by a reliable secondary source was deemed sufficient recommendation for current purposes.

The inadequacy of information on the influence of temperature and of moisture content constitutes a general limitation upon the data presented. For some materials data on the effects of *temperature* were found; in these

cases data for the range of temperatures encountered in air conditioning practice are included in the tables. Data on the effect of *moisture content* were found only for wood and a few isolated masonry materials; the other entries are for either dry materials or unspecified (presumably average) moisture contents.

It is important that the units used be defined. Data are presented in units

TABLE 1—THERMO-PHYSICAL PROPERTIES OF WOOD

WOOD	k	ρ	$k\rho c_p$	THERMAL RESISTANCE PER INCH THICKNESS	$a = \frac{k}{\rho c_p}$
	(9% Moisture)	(9% Moisture)	(80 F, 9% Moisture)[a]	(9% Moisture)	(80 F, 9% Moisture)[a]
	Btu / (Hr) (Sq Ft) (°F/Ft)	Lb / (Cu Ft)	$(Btu)^2$ / $(Hr)(Ft)^4(°F)^2$	$\frac{1}{12k}$	(Sq Ft)/(Hr)
Hardwoods					
Ash, white	0.098	38.2	1.45	0.85	0.0065
Beech	0.10	39.7	1.53	0.84	0.0065
Birch, yellow	0.098	39.1	1.48	0.85	0.0065
Chestnut	0.073	27.1	0.76	1.14	0.0070
Elm, American	0.083	32.6	1.04	1.01	0.0066
Mahogany[b]	0.075[b]	34.3[b]	1.30	1.11[b]	0.0057[b]
Maple, silver	0.079	30.5	0.93	1.05	0.0067
Maple, sugar	0.10	40.2	1.56	0.84	0.0064
Oak, northern red (commercial)	0.10	39.3	1.52	0.84	0.0066
Oak, white (commercial)	0.105	42.1	1.71	0.80	0.0065
Sweetgum	0.079	31.4	0.96	0.93	0.0065
Softwoods					
Baldcypress	0.076	28.8	0.85	1.05	0.0068
Balsa[b]	0.048[b]	20.0[b]	0.37	1.72[b]	0.0062
Balsa[b]	0.032[b]	8.8[b]	0.11	2.63[b]	0.0094
Balsa[b]	0.028[b]	7.3[b]	0.079	3.03[b]	0.0099
Douglas fir—coast type	0.081	30.7	0.96	1.03	0.0068
Fir, grand	0.068	25.3	0.64	1.22	0.0070
Hemlock, eastern	0.070	25.9	0.70	1.19	0.0070
Pine, loblolly	0.085	32.4	1.06	0.99	0.0069
Pine, lodgepole	0.070	25.9	0.70	1.19	0.0070
Pine, longleaf	0.096	37.2	1.38	0.87	0.0067
Pine, red	0.079	30.5	0.93	0.93	0.0067
Pine, ponderosa	0.070	25.5	0.69	1.19	0.0071
Pine, shortleaf	0.083	32.2	1.03	1.01	0.0067
Redwood	0.070	25.5	0.69	1.19	0.0071
Spruce, Engelmann	0.059	21.1	0.48	1.42	0.0072
Spruce, Sitka	0.068	25.3	0.66	1.22	0.0070

[a] c_p = 0.386 Btu per (pound) (Fahrenheit degree). Moisture is percent of oven-dry weight of wood.
[b] HEATING, VENTILATING, AIR CONDITIONING GUIDE 1947, moisture not stated.
Note: k and ρ for wood are not appreciably influenced by temperature variations in the range of ordinary air conditioning applications. c_p is only very slightly dependent on temperature, increasing 0.000644 units per Fahrenheit degree increase in temperature. All three properties are significantly influenced by changes in moisture content, but moisture content is very uncertain to predict with assurance. A moisture content of 9 percent is recommended for average summer conditions in outside building walls.

consistent with the Mackey and Wright design charts.[1] Thermal conductivities are expressed in Btu per (hour) (square foot) (Fahrenheit degree per foot). This is a departure from the current tables in the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1947 where the temperature gradient in Fahrenheit degrees per inch is employed. In recognition of the familiarity attached to the

inch unit, all magnitudes of the thermal resistance have been tabulated in the same units used in THE GUIDE, that is, per inch of material thickness.

VI. DATA ON WOOD

A. Scope

The properties under consideration are influenced predominantly by species, moisture content, and temperature. Only domestic woods, over the range of moisture content and temperature to be expected in practice, were covered in this study.

The individual species of wood are characterized, as far as their thermo-

TABLE 2—SPECIFIC GRAVITIES OF WOOD

WOOD	S_g	S_d
Hardwoods		
Ash, white	0.55	0.64
Beech	0.56	0.67
Birch, yellow	0.55	0.66
Chestnut	0.40	0.45
Elm, American	0.46	0.55
Maple, silver	0.44	0.51
Maple, sugar	0.56	0.68
Oak, northern red (commercial)	0.56	0.66
Oak, white (commercial)	0.59	0.71
Sweetgum	0.44	0.53
Softwoods		
Baldcypress	0.42	0.48
Douglas fir (coast type)	0.45	0.51
Fir, grand	0.37	0.42
Hemlock, eastern	0.38	0.43
Hemlock, western	0.38	0.44
Larch, western	0.48	0.59
Pine, loblolly	0.47	0.54
Pine, lodgepole	0.38	0.43
Pine, longleaf	0.54	0.62
Pine, red	0.44	0.51
Pine, ponderosa	0.38	0.42
Pine, shortleaf	0.46	0.54
Redwood	0.38	0.42
Spruce, Englemann	0.31	0.35
Spruce, Sitka	0.37	0.42

S_g = Specific gravity based on volume when green and weight when oven-dry.
S_d = Specific gravity based on volume and weight when oven-dry.

physical properties are concerned, by the values of specific gravity when green and when oven-dried. That is, if these two specific gravities are known, the desired properties may be determined for any species from information summarized in this Bulletin.

The data can be considered reliable since they have been obtained from the *U. S. Forest Products Laboratory.*

The complete set of curves for wood, Figs. 1 through 7, inclusive, is intended for general reference purposes. Table 1 presents a quick reference summary for practical use.

B. The Equilibrium Moisture Content of Wood

Fig. 1 presents equilibrium moisture-content data.[3] These data will serve as the basis of practical estimates for all of the woods listed in Table 2 under atmospheric exposure and in an untreated condition.

A survey conducted by the *U. S. Forest Products Laboratory* showed that on the shaded side of a dwelling the average moisture content of the sheathing during summer months was about 9 percent and that of the siding about 10 percent. On the sunny side, the moisture content was about 2 percent lower.

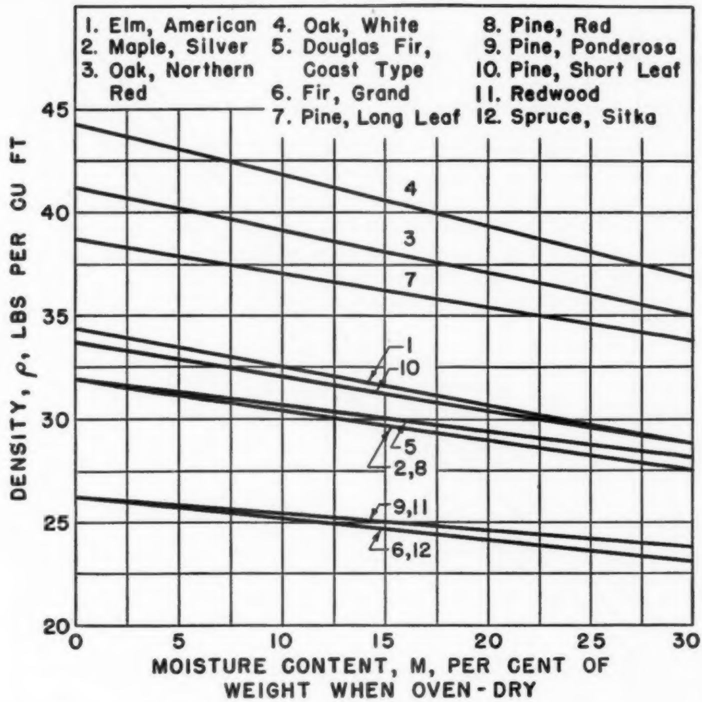


FIG. 2. RELATION BETWEEN DENSITY AND MOISTURE CONTENT FOR REPRESENTATIVE WOODS

Variations from these figures may be expected with extremes of climate and locality, but these magnitudes will be a general guide.

C. The Specific Gravity and Density of Wood

Considerable data are available on the specific gravities of common woods.[3] Whenever specific gravity is to be stated for wood, it is necessary to give the moisture content and basis of volume measurement corresponding to the specific gravity. The moisture content may vary from 30 percent of the oven-dry weight of the wood when green (fiber saturation point) to zero when dried. Wood, furthermore, undergoes a very appreciable expansion or contraction when taking on or giving up moisture.

The specific gravity of wood at any intermediate moisture content between

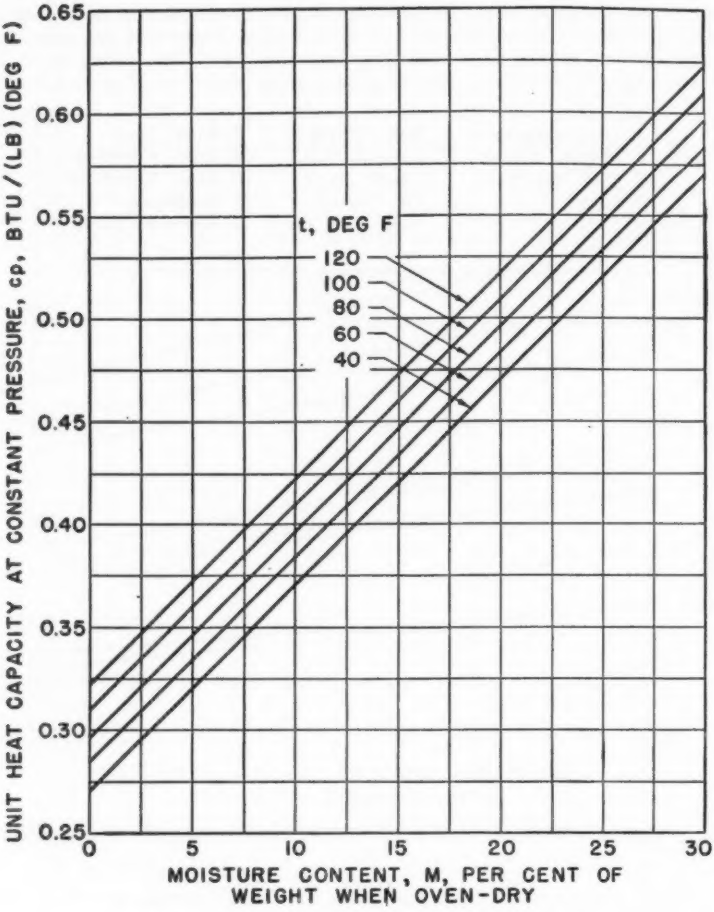


FIG. 3. UNIT HEAT CAPACITY OF WOOD AT CONSTANT PRESSURE

green and oven-dry may be calculated from known magnitudes of two specific gravities, S_g and S_d:

S_g = specific gravity based on the volume when green and the weight when oven-dry.
S_d = specific gravity based on the volume when oven-dry and the weight when oven-dry.

For any moisture content, M, expressed as a percentage of the oven-dry weight, the specific gravity, S, will be:

$$S = S_d - (S_d - S_g)\frac{M}{30} \quad \ldots \quad (1)$$

This relation is based on the recommended working approximation that volume shrinkage is proportional to the reduction in moisture content.[3] Densities in pounds per cubic foot are obtained by multiplying specific gravities by 62.4.

Table 2 presents recommended average data, applicable at all ordinary temperatures, on S_d and S_g for 25 species of trees. The oven-dry weight of wood is obtained after drying a sample in an oven at a temperature between 212 and 221 F until the weight becomes constant. All volatile matter driven off in this drying process can be considered as water, *i.e.,* moisture. The *U. S. Forest Products Laboratory* states that knots, grain contours, and local growing conditions have negligible influence upon the physical properties of wood.

Fig. 2 presents the relation between density and moisture content for a few representative woods, as determined from Equation 1 and the data of Table 2. The slope of the curves may be opposite to first expectations, but the decrease of density with increasing moisture content is a consequence of the very appreciable swelling undergone by common woods in taking up moisture.

D. The Unit Heat Capacity‡ of Wood

The results of tests to determine unit heat capacity, c'_p, of oven-dried wood at a constant pressure of one atmosphere are represented[4] by the equation:

$$c'_p = 0.245 + 0.000644t, \text{ Btu per (pound) (Fahrenheit degree)} \quad \ldots \quad (2)$$

where

t = wood temperature, Fahrenheit degrees.

This relation is applicable to all species with good accuracy.

The effect of moisture must be added to Equation 2 to meet the conditions of field problems. Since c_p is unity for water,¶ the unit heat capacity for moist wood may be written:

$$c_p = 0.245 + 0.000644t + \frac{M}{100}, \text{Btu per (pound) (Fahrenheit degree)} \quad \ldots \quad (3)$$

Fig. 3 shows this relation graphically. No data are available to verify directly the influence of moisture upon the unit heat capacity. Direct addition of the moisture term is in accord with the recommendations of the *U. S. Forest Products Laboratory.*

E. The Thermal Conductivity of Wood

The *U. S. Forest Products Laboratory* has conducted extensive research on the thermal conductivity of wood. A good practical correlation of the results is expressed by the equation:

$$k = \frac{S_g(0.116 + 0.0023M)}{1 - S_g[0.009(30 - M)]} + 0.01375 \text{ Btu per (hour) (square foot) (Fahrenheit degree per foot)} \quad \ldots \quad (4)$$

‡ Throughout this Bulletin, the unit heat capacity is considered at constant pressure.
¶ It is believed that intracellular water in wood may be in a somewhat different state than free water,[1] but this influence upon its unit heat capacity is not of practical significance.

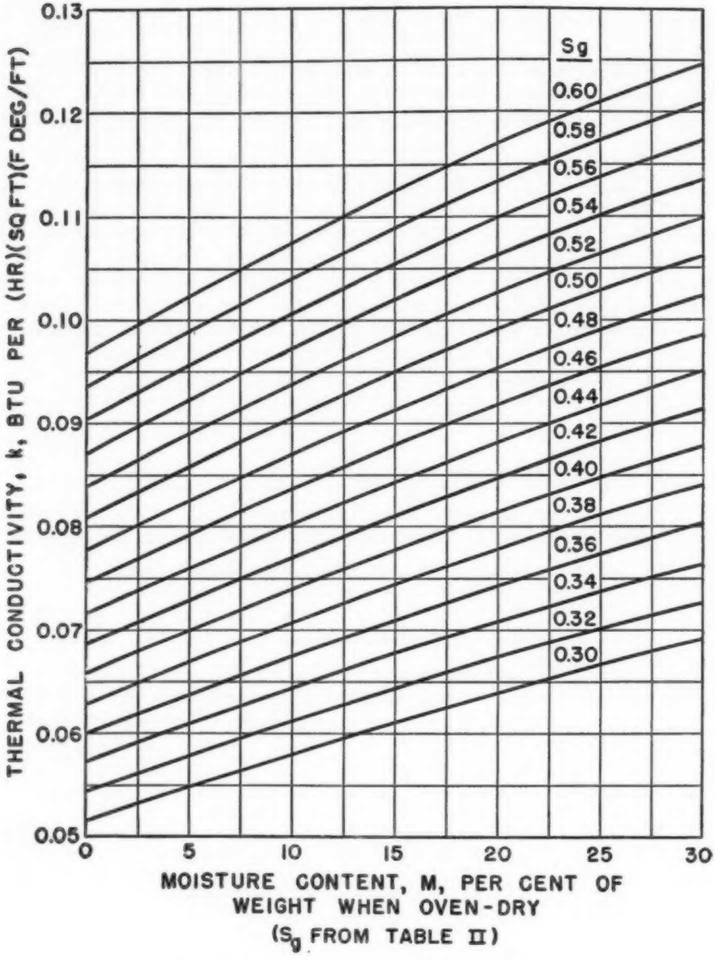


FIG. 4. THERMAL CONDUCTIVITY OF WOOD

Fig. 4 shows this relation graphically. The conductivity magnitudes serve for cross-grain heat flow in all species below the fiber saturation point.

The influence of temperature is reported [5] to cause only a negligible increase of thermal conductivity with temperature rise in the range of interest.

Application of a temperature gradient to moist wood will cause a slight drift of apparent mean thermal conductivity with time, since moisture diffuses

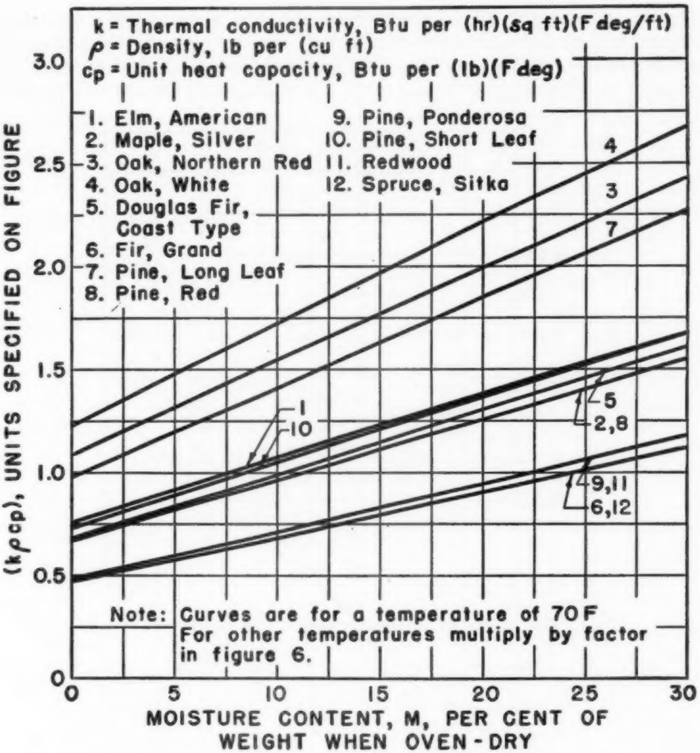


FIG. 5. MAGNITUDES OF ($k\rho c_p$) FOR REPRESENTATIVE WOODS

through the wood as a result of a non-equilibrium concentration under the influence of a temperature gradient. This effect usually may be neglected for the purposes of air conditioning load estimates.

F. The Product ($k\rho c_p$) and Thermal Diffusivity

Magnitudes of the product ($k\rho c_p$) are presented in Fig. 5 as a function of moisture content for the representative woods having density curves in Fig. 2. These data are plotted for a temperature of 70 F. The effect of a temperature different from 70 F may be readily determined by multiplying the values from Fig. 5 by the proper correction factor from Fig. 6. These latter correction factors account for the influence of temperature upon the unit heat capacity (Fig. 3).

The product ($k\rho c_p$) is needed in the use of the Mackey and Wright design charts for periodic heat flow [1] in which it is employed as a basic factor.

Thermal diffusivities are shown in Fig. 7 for the same representative woods as shown in Fig. 5. It is interesting to note that the spread of the curves for the different species is less in Fig. 7 than in Fig. 5, indicating that the thermal diffusivity may be preferable to $(k\rho c_p)$ for calculations should the properties

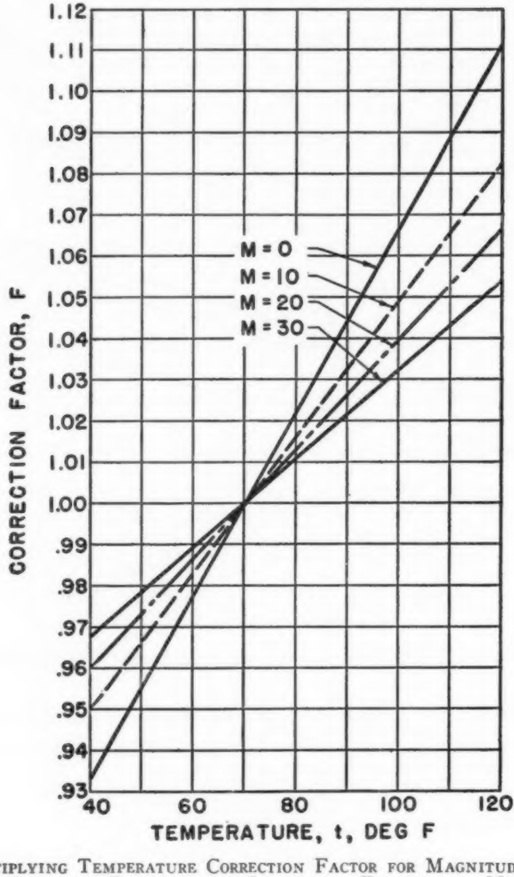


FIG. 6. MULTIPLYING TEMPERATURE CORRECTION FACTOR FOR MAGNITUDES OF $(k\rho c_p)$ IN FIG. 5. DIVIDING TEMPERATURE CORRECTION FACTOR FOR MAGNITUDES OF $a = k/\rho c_p$ IN FIG. 7

of a particular wood be uncertain. The temperature correction is obtained in this instance by dividing the diffusivity by the factor from Fig. 6.

G. A Practical Summary Table for Wood

In considering the preparation of a practical summary table a compromise is necessary because the properties cannot be tabulated for the entire range of

moisture contents and temperatures without so complicating and expanding the table that it becomes impractical. It has been noted that the unit heat capacity is the only property for which a variation with temperature has been estab-

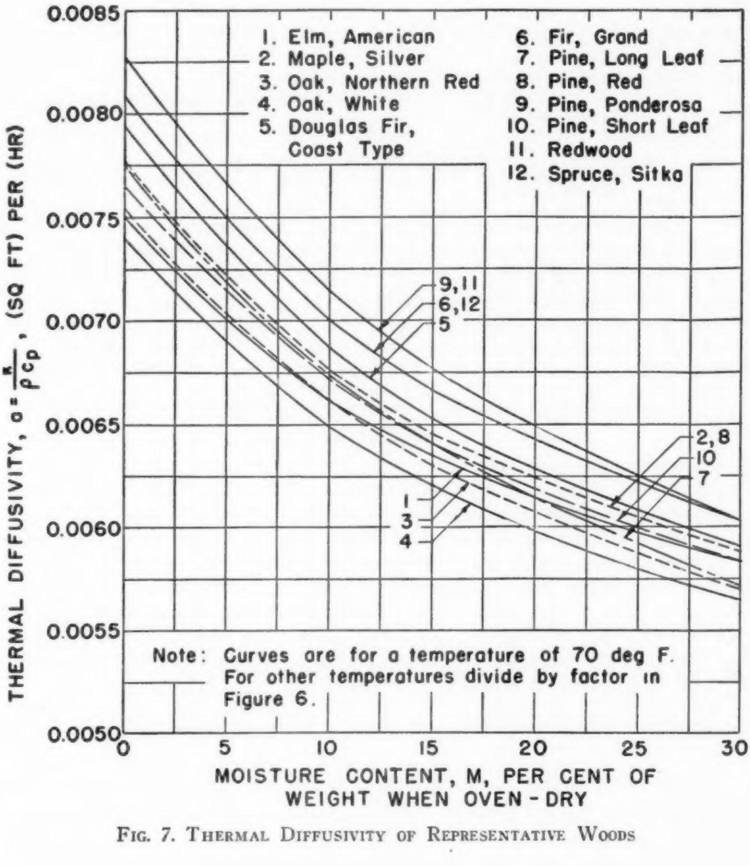


FIG. 7. THERMAL DIFFUSIVITY OF REPRESENTATIVE WOODS

lished. Thermal conductivity and density have been considered to be uninfluenced by temperature for practical purposes in the range of ordinary air conditioning applications. Moisture content has a significant effect upon all three properties; but moisture content of building materials is a variable and uncertain item. As the moisture survey mentioned previously indicates that a choice of 9 percent moisture for outer building walls is reasonable, this value has been selected as the basis of the properties to be tabulated. A choice of 80 F as the reference temperature is suggested since the periodic heat-flow

problem belongs largely with summertime conditions. Table 1 presents the suggested practical summary for wood.

H. Symbols

Symbols appearing in Tables 1-7 are as follows:

a = thermal diffusivity, square feet per hour.
c_p = unit heat capacity of moist wood at constant pressure, Btu per (pound) (Fahrenheit degree).
c'_p = unit heat capacity of oven-dried wood at constant pressure, Btu per (pound) (Fahrenheit degree).
k = thermal conductivity of wood, Btu per (hour) (square foot) (Fahrenheit degrees per foot).
M = moisture content of wood, percent of weight when oven-dry.
S = specific gravity of moist wood at any stated condition.
S_d = specific gravity of wood, based on volume and weight when oven-dry.
S_g = specific gravity of wood, based on volume when green and weight when oven-dry.
t = temperature, Fahrenheit degrees.
ρ = density, pounds per cubic foot.

VII. DATA ON BUILDING AND INSULATING BOARDS AND INSULATING MATERIALS

A. Scope

Data on these materials as obtained from the literature are quite satisfactory for products now in general use. The values have been limited to dry materials as moisture is reported to produce uncertain effects. The nature of some of the materials makes possible appreciable variations in bulk density and this, in turn, gives rise to variations in thermal conductivity. Where available, an approximate range of variation in these properties has therefore been recorded. The average data reported for density and thermal conductivity will be sufficient for most practical application calculations.

Table 3 gives a practical summary of the available data. The references consulted are indicated in column 2 and are listed on pages 34-35.

B. Comments on the Data

Adequate data on the influence of temperature upon thermo-physical properties are available for only a part of the materials listed. Where this influence is known, the properties have been given at three temperatures. (The references indicated should be consulted for complete data on the temperature variation.) Where the information was found incomplete, the temperatures corresponding to the data listed have been noted in the table.

The unit heat capacities generally fall within the range expected for materials of vegetable and mineral origin. This type of material has an appreciably temperature-sensitive value of unit heat capacity [6, 7] which is evidenced by the difference between the true and mean values of c_p, where both are given in Table 3.

The density and thermal conductivity figures demonstrate the customary trend

of low density corresponding to low thermal conductivity, and vice versa. This is further accentuated in the magnitudes of $(k\rho c_p)$.

The values of $(k\rho c_p)$ have been tabulated on the basis of the known data, with no additional temperature adjustment when data at a common temperature were lacking. This is acknowledged as a compromise and was necessary because the temperature dependence of each constituent factor was not always available. However, since both the data and the usual practical application for which the data are intended do not ordinarily warrant more than two significant figures, the compromise is considered to be justified.

VIII. DATA ON MASONRY MATERIALS

A. Scope

Table 4 lists the properties of seven common masonry materials. Since all masonry materials in this classification are either natural mineral products or processed aggregations of natural materials, a range of variation must be expected in their properties, according to differences in the natural deposits from which they are obtained and to differences in processing. The data of Table 4 present both average magnitudes of the properties and ranges of expected variation as far as these are known. (It is advisable to consult the original references for further details on materials from particular geographical locations, particularly reference number 17.)

Moisture content and temperature have an influence on the properties of masonry materials, but the available information is entirely too meager to permit generalization. Limited data for moist brickwork and sandstone are quoted in Table 4.

B. Comments on the Data

Since all masonry materials are of mineral origin, the unit heat capacities are grouped around the value of 0.2 Btu per (pound) (Fahrenheit degree).

The influence of density is very evident in the data in Table 4. The densities of all natural rock-type materials vary approximately as the ***hardness***. As building brick and tile are made from clay or shale and are appreciably porous, their densities are considerably less than the natural rock types of material. Also, brick and tile densities are subject to considerable variation according to the method of manufacture employed.

As would be expected, the trend in magnitudes of thermal conductivity follows that for density. The range in variation of conductivity is relatively greater than the corresponding range of density, however, and illustrates the sensitive dependence of the rate of heat conduction upon the compactness of a given type of material.

The very considerable increase of thermal conductivity with moisture content for brick and sandstone (the only two materials for which data could be found) is a striking demonstration of the effect produced when minute air spaces are filled with water.

A mean value of thermal conductivity is usually satisfactory for practical air conditioning load estimates. The data of Table 4, combined with good judgment, should meet all ordinary needs.

TABLE 3—THERMO-PHYSICAL PROPERTIES OF BUILDING AND INSULATING BOARDS AND INSULATING MATERIALS[a]

MATERIAL	REFERENCE[b]	c_p UNIT HEAT CAPACITY Btu/(Lb)(F Deg)				ρ DENSITY (Lb)/(Cu Ft) ORDINARY TEMPS		k THERMAL CONDUCTIVITY Btu / (Hr)(Sq Ft)(F Deg/Ft)		$k\rho c_p$ $\frac{(Btu)^2}{(Hr)(Ft)^4(F\ Deg)^2}$	THERMAL RESISTANCE PER INCH THICKNESS $\frac{1}{12k}$	$a = \frac{k}{\rho c_p}$ THERMAL DIFFUSIVITY (Sq Ft)/Hr
		True	At Temp F	Mean	Over Temp Range F	Variation Range	Avg	k	Mean Temp F			
Asbestos Paper, Corrugated	7			0.245	80–212	12–22	17	0.037	150	0.15	2.3	0.0089
Asbestos	10, 20			0.195	68–208		29.3	0.090	80	0.51	0.93	0.016
Asbestos							36	0.111	212	0.78	0.75	0.016
Asbestos Board, Corrugated	11			0.2			20	0.040	110	0.16	2.1	0.010
Asbestos Millboard, Pressed	10,11,12,13			0.2			60	0.070	86	0.84	1.2	0.0058
Cement and Asbestos Sheets, Pressed	11,12,13,16			0.2			123	0.225	86	5.5	0.37	0.0091
Corkboard, Average	7, 6, 19	0.40 0.37 0.34	120 80 40	0.392	76–142		6.9	0.023 0.022 0.020	120 80 40	0.056	3.8	0.0086
Glass Wool	7, 11, 24	0.205 0.198 0.190	120 80 40	0.21	80–212	1.5–4.5	3	0.023	75	0.013	3.7	0.038
Glass Wool with Binder	7	0.189 0.179 0.169	120 80 40	0.196	80–212		12	0.025	150	0.054	3.3	0.012
Gypsum Board	11, 12			0.26	60–105	53–63	58	0.12	70	1.8	0.71	0.0080
Gypsum, Cellular	20			0.26			30	0.083	70	0.65	1.0	0.011
Expanded Glass Block	6, 19	0.180 0.174 0.167	120 80 40	0.179	79–144	9.5–11.5	10.5	0.036 0.035 0.034	120 80 40	0.064	2.4	0.019
Hair Felt	7, 20, 24			0.334	80–212		11	0.0233 0.0217 0.0201	120 80 40	0.080	3.8	0.0059
Kapok	10, 16	0.324	68				9.4	0.020–0.023	68	0.065	3.9	0.0071
Lead Slag Wool	7	0.175 0.170 0.164	120 80 40	0.178	80–212	4–10	7	0.022	150	0.026	3.8	0.019
85 Percent Magnesia	7	0.274 0.272 0.270	120 80 40	0.276	80–212	12–18	15	0.035	150	0.14	2.4	0.0086

Redwood Bark, Shredded	6, 19	0.244 0.239 0.234	120 80 40	0.246	78–139	3–5	4	0.025 0.023 0.021	120 80 40	0.022	3.6	0.024
Rock Wool	7	0.198 0.195 0.191	120 80 40	0.201	80–212	2–10	6	0.023	150	0.027	3.6	0.020
Rock Wool with Asphaltic Binder	7			0.247	80–212		17	0.028	150	0.12	3.0	0.0067
Vermiculite	7, 11	0.198 0.189 0.180	120 80 40	0.205	80–212	8–10	9	0.045	150	0.077	1.9	0.026
Vermiculite							6.2	0.027		0.032	3.1	0.023
Vermiculite Board	24			0.21			18.9	0.0788 0.0767 0.0745	120 80 40	0.30	1.1	0.0019
Wood Fiber Blanket	7			0.330	80–212	2–4	3	0.023	150	0.023	3.6	0.023
Glass Fiber Board	6	0.235 0.230 0.220	120 80 40	0.236	79–142		11	0.0235 0.022 0.020	120 80 40	0.056	3.8	0.0087
Mineral Wool Board	6, 19, 24	0.190 0.185 0.180	120 80 40	0.190	78–143		14.3	0.026 0.0245 0.023	120 80 40	0.063	3.4	0.0093
Expanded Rubber Board	6	0.278 0.255 0.232	120 80 40	0.273	78–144		4.9	0.019 0.0185 0.018	120 80 40	0.022	4.6	0.015
Vegetable Fiber Board	6, 19	0.284 0.265 0.246	120 80 40	0.279	77–141		14.4	0.0285 0.0275 0.0265	120 80 40	0.10	3.0	0.0072
Wood Fiber Board, Dry	7, 16, 24			0.341	80–212	12–17	15.7	0.0301 0.0287 0.0272	120 80 40	0.15	2.9	0.0054
Wood Fiber Board, 24% Moisture	24			0.4			15.7	0.0367 0.0349 0.0331	120 80 40	0.22	2.4	0.0056

[a] Tabulated magnitudes of $(k\rho c_p)$, a, and $1/12k$ are based upon the following values:
k as listed, 80 F or mean value where range given.
ρ average listed.
c_p true where listed at 80 F, otherwise mean.
No adjustment has been made to bring values of k, ρ, and c_p to the same temperature.

[b] See list of references on pages 34–35.

TABLE 4—THERMO-PHYSICAL PROPERTIES OF MASONRY MATERIALS[a]

MATERIAL	REFERENCE[b]	c_p UNIT HEAT CAPACITY Btu/(Lb)(F Deg)				ρ DENSITY (Lb)/(Cu Ft) ORDINARY TEMPS		k THERMAL CONDUCTIVITY Btu / (Hr)(Sq Ft)(F Deg/Ft)		$k\rho c_p$ $\frac{(\text{Btu})^2}{(\text{Hr})(\text{Ft})^4(\text{F Deg})^2}$	THERMAL RESISTANCE PER INCH THICKNESS $\frac{1}{12k}$	$a = \frac{k}{\rho c_p}$ THERMAL DIFFUSIVITY (Sq Ft)/Hr
		True	At Temp F	Mean	Over Temp Range F	Variation Range	Avg	k	Mean Temp F			
Basalt (Lava Rock)	10, 21, 23, 17			0.205	32–210	169–200	184	0.74–1.61	121	44.	0.11–0.052	0.031
Building Brick and Tile Material:												
Soft, Dry	23			0.18	32–175		105	0.30	68	5.7	0.28	0.016
Soft, 1.1% Moisture by Weight	23			0.19	32–175		106	0.55	68	11.	0.15	0.027
Medium	21, 22, 25			0.195–0.220			112	0.33–0.42	70	8.7	0.25–0.20	0.016
Hard				0.195–0.220			128	0.33–0.42		9.9	0.25–0.20	0.014
Common Yellow Clay Brick	11, 12			0.22			115	0.40		10.	0.21	0.016
Common Yellow Clay Brick, Damp or Wet	11							0.42				
Brickwork, 4½ Months Old	10			0.2			110	0.56	50	12.	0.15	0.025
Brickwork, Very Old and Dry	10, 23			0.2			116	0.24	68	5.6	0.35	0.010
Brickwork, New and Moist	10			0.2			122	0.81	50	20.	0.10	0.033
Granite	10, 12, 17, 21, 23			0.20	68–210	156–187	172	1.8–2.4		72.	0.046–0.035	0.061
Limestone	10, 12, 17, 21, 23			0.22	59–210	144–169	156	0.73	210	25.	0.11	0.021
Marble	10, 12, 17, 21, 22, 23			0.21	77–210	162–178	171	1.2–1.9		56.	0.070–0.044	0.043
Porcelain (Tile)	23			0.26		144–156	150	0.33		13.	0.25	0.0085
Sandstone, Dry	10, 21, 17			0.18		137–162	150	0.75	68	20.	0.11	0.028
Sandstone, Moist				0.2		137–162	150	0.97	68	29.	0.086	0.032

[a] Tabulated magnitudes of $(k\rho c_p)$, a, and $1/12k$ are based upon the following values:
k as listed, or average for $(k\rho c_p)$ and a, where range given.
ρ average listed.
c_p mean listed, or average where range given.
No adjustment has been made to bring values of k, ρ, and c_p to the same temperature.

[b] See list of references on pages 34–35.

IX. DATA ON CONCRETE, PLASTER, AND MORTAR

A. Scope

The materials of this classification are mixtures made from different natural and prepared constituents. Both the nature and proportions of the constituents are subject to considerable variations.

The materials listed in Table 5 have been chosen to typify ordinary structural uses. These are: (1) sand and gravel concrete, (2) various lightweight aggregate concretes, (3) cement mortar, (4) gypsum and lime-cement plasters, (5) stucco, and (6) gypsum fiber concrete.

B. Comments on the Data

A few additional lightweight concretes, for which the density and thermal conductivity are listed in THE GUIDE 1947, may be assigned a value of the unit heat capacity, c_p, equal to 0.21 Btu per (pound) (Fahrenheit degree). Accordingly, the essential data on their properties may be considered complete for practical purposes.

The magnitudes of the unit heat capacities listed in Table 5 vary only slightly for the different entries. This agrees with the fact that all mineral aggregate concretes contain the same types of chemical substances. Since these are only average data, the fact that concrete is a mixture explains the variations of 10 or 15 percent from the figures listed at any fixed temperature and moisture content.

Density variations in concrete reflect differences in the aggregate and the porosity of the mix after setting. Variations in thermal conductivity naturally follow the variations in density.

Cement mortars and plasters show no significant differences in properties from plain concrete as they are similar in nature. Lime is frequently added to improve their workability but has no significant effect upon their thermophysical properties. Stucco is a special type of mortar.

Gypsum plasters and concretes show differences in the matter of thermophysical properties in comparison to the cement materials. The introduction of gypsum leads to a slight increase in the unit heat capacity, a lower density, and a decreased thermal conductivity.

Table 5 shows a wide range for magnitudes for the product $(k\rho c_p)$. This range is pertinent in considering the design of structures from a purely thermal standpoint.

X. DATA ON GLASS

A. Scope

Glass is a highly adaptable material which exists in many different compositions [8] and which has many uses. Window and structural glass in ordinary use for buildings, however, is generally restricted to the following types: (1) soda-lime-silica glass for plate, windows, and structural blocks, (2) opaque colored glass for wall finishing and partitions, (3) laminated safety glass,

TABLE 5—THERMO-PHYSICAL PROPERTIES OF STRUCTURAL CONCRETE, PLASTER, AND MORTAR[a]

MATERIAL (DRY)	REFERENCE[b]	c_p UNIT HEAT CAPACITY Btu/(Lb)(F Deg)				ρ DENSITY (Lb)/(Cu Ft) ORDINARY TEMPS		k THERMAL CONDUCTIVITY Btu / (Hr)(Sq Ft)(F Deg/Ft)			$k\rho c_p$ $\frac{(Btu)^2}{(Hr)(Ft)^4(F\ Deg)^2}$	THERMAL RESISTANCE PER INCH THICKNESS $\frac{1}{12k}$	$a = \frac{k}{\rho c_p}$ THERMAL DIFFUSIVITY (Sq Ft)/Hr
		True	At Temp F	Mean	Over Temp Range F	Variation Range	Avg	Variation Range	Avg	Mean Temp F			
Neat Cement	18, 23			0.28	212–392		114.		0.356	302	11.4	0.23	0.011
Concrete[c]													
Sand and Gravel or Other Heavy Natural Aggregate, Mixes 1:2–1:9, Relative Water 110%	18			0.22	212–392		142.	0.78–0.93	0.86	302	27.	0.097	0.028
Sand and Gravel Aggregate, Various Ages and Mixes	26, 11			0.22		140–150	145.	0.95–1.36	1.15		37.	0.072	0.036
Cinder Aggregate, Structural	26, 11			0.21		89–118	97.		0.41	75	8.4	0.20	0.020
Expanded Burned Clay Aggregate	26, 11												
Structural				0.21		90–110	100.	0.186–0.333	0.332		7.0	0.30	0.016
Floor Fill				0.21		70–80	74.	0.135–0.311	0.217		3.4	0.38	0.014
Roof Fill				0.21		60–75	73.	0.133–0.190	0.135		2.1	0.62	0.0088
Expanded Slag Aggregate	26												
Structural				0.21		100–110	100.		0.317		6.7	0.26	0.015
Insulating Fill				0.21		60–85	76.	0.125–0.150	0.133		2.1	0.63	0.0083
Gypsum Plaster	11, 27, 21			0.23			70.		0.275		4.4	0.30	0.017
Lime-Cement Plaster	11, 23, 14			0.22		103–117	110.		0.67		16.	0.125	0.028
Cement Mortar	11, 14, 21, 23			0.22		94–135	115.		1.0		25.	0.083	0.040
Stucco	11			0.22			110.		1.05		25.	0.079	0.043
Gypsum Fiber Concrete, 87½% Gypsum 12½% Wood Chips	11			0.27			51.		0.138	74	1.9	0.60	0.010

[a] Tabulated magnitudes of ($k\rho c_p$), $1/12k$ and a are based upon the following values:
k average listed.
ρ average listed.
c_p mean listed.
No adjustment has been made to bring values of k, ρ, and c_p to the same temperature.

[b] See references on pages 34–35.

[c] For other special concretes, use present HEATING, VENTILATING, AIR CONDITIONING GUIDE Data for k and ρ, and assume c_p = 0.21.

(4) plate glass heat-treated for high strength, and (5) heat-absorbing plate glass. Table 6 notes their thermo-physical properties.

B. Comments on the Data

The properties of glass are dependent upon its chemical composition and the treatment received during the manufacturing process. References 8 and 9 should be consulted for complete information.

The ordinary soda-lime-silica glass most commonly used in buildings generally has compositions by weight in the following range:

Silica (silicon dioxide)	68 to 75 percent
Lime (calcium oxide)	10 to 14 percent
Soda (sodium oxide)	11 to 15 percent

These limits of composition are by no means rigorous, and traces of other materials are usually found. Air conditioning load estimates are primarily concerned with this type of glass in its various forms, including special treatment in manufacture or the use of additives to produce desired variations in properties.

Laminated safety glass and high-strength heat-treated plate glass have essentially the same thermo-physical properties as the parent soda-lime-silica glass, and only their mechanical properties differ. Heat-absorbing plate glass is produced by the addition of a very small amount (roughly 0.6 percent) of ferric oxide to the ordinary mix, in which the iron is converted during manufacture to the ferrous state to serve as the absorbing agent for the infra-red portion of incident radiation. The only thermo-physical property affected appreciably in this special glass is the unit heat capacity which is slightly increased in magnitude.

The remaining type of structural glass listed in Table 6 is the opaque glass employed for wall finishing, partitions, and decorative purposes. Black opaque glass is ordinary soda-lime-silica glass to which manganese and chromium have been added in considerable amounts as coloring agents.

All other colors of opaque glass belong to the soda-alumina-silica family of glasses in which aluminum oxide replaces a large portion of the lime. The opaque character of the glass is created by the addition of sodium-silico-fluoride to the melt. This latter compound produces a milky effect within the body of the glass during annealing, and the various colors are obtained by the addition of mineral coloring agents. The thermo-physical properties of these opaque glasses differ somewhat from clear glass, as noted in Table 6. Transparent plate glass is also produced in several tinted colors, but the tinting does not noticeably affect its thermo-physical properties.

Since glass is a mineral substance, it may be noted that the individual properties of thermal conductivity, density, and unit heat capacity are of the same order of magnitude as for other mineral materials, such as listed in Table 4 for masonry.

XI. DATA ON ROOFING MATERIALS

A. Scope

The most common roofing materials are covered by Table 7. Various built-up felt-base roofs and prepared asphalt-base shingles represent the ma-

TABLE 6—THERMO-PHYSICAL PROPERTIES OF GLASS USED IN BUILDINGS[a]

MATERIAL	REFERENCE[b]	c_p UNIT HEAT CAPACITY Btu/(Lb)(F Deg)				ρ DENSITY (Lb)/(Cu Ft) ORDINARY TEMPS AVG	k THERMAL CONDUCTIVITY $\frac{Btu}{(Hr)(Sq\ Ft)(F\ Deg/Ft)}$		$k\rho c_p$ $\frac{(Btu)^2}{(Hr)(Ft)^4(F\ Deg)^2}$	THERMAL RESISTANCE PER INCH THICKNESS $\frac{1}{12k}$	$a = \frac{k}{\rho c_p}$ THERMAL DIFFUSIVITY (Sq Ft)/Hr
		True	At Temp F	Mean	Over Temp Range F		k	Mean Temp F			
Ordinary Soda-Lime-Silica Glass for Plate, Windows, and Building Blocks[c]	30, 31			0.205	32–212	157	0.556	120	18.	0.150	0.017
Opaque Colored Glass for Partitions and Wall Finishing—	30										
Black				0.207	32–212	158	0.627	120	20.5	0.133	0.019
Other Colors				0.202	32–212	152	0.701	120	21.5	0.119	0.023
Laminated Safety Glass Made from Ordinary Plate Glass	30			Same	as for	Plate Glass					
Regular Plate Glass Heat-Treated for High Strength	30			Same	as for	Plate Glass					
Heat-Absorbing Plate Glass	30, 31			0.22	32–212	157	0.556	120	19.	0.150	0.016

[a] Tabulated magnitudes of ($k\rho c_p$), $1/12k$, and a are based upon the following values:
k as listed.
ρ as listed.
c_p mean listed.
No adjustment has been made to bring values of k, ρ, and c_p to the same temperature.

[b] See references on pages 34–35.

[c] Approximate range of composition:
SiO_2, Silica, 68–75%
CaO, Lime, 10–14%
Na_2O, Soda, 11–15%

jority of present-day roof constructions. Although the available data were found to be rather meager, they are believed to be adequate for design estimates in view of the relatively standardized nature of roofing constructions.

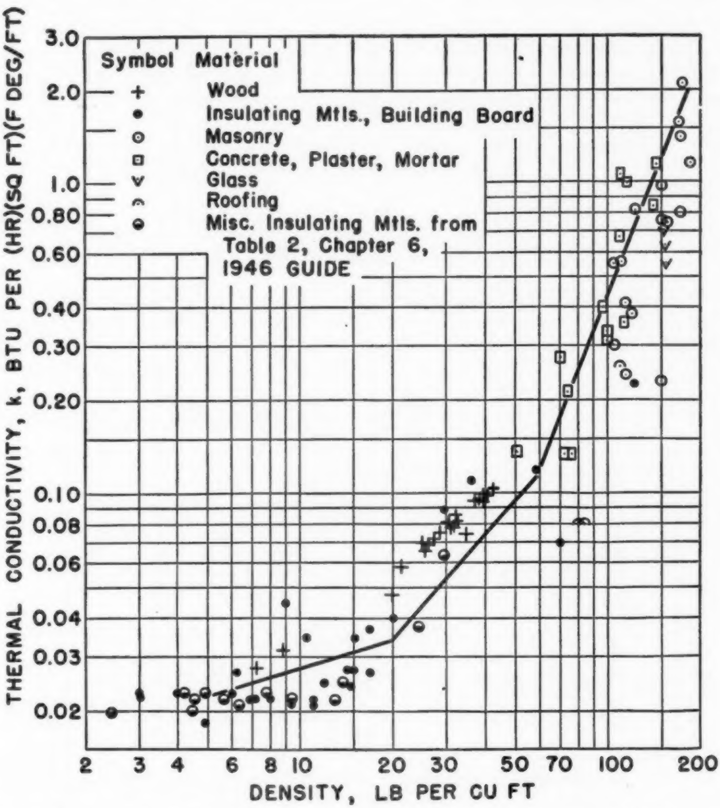


FIG. 8. COMPARISON BETWEEN DATA OF THIS BULLETIN AND EMPIRICAL ESTIMATING EQUATIONS FOR THERMAL CONDUCTIVITY

No quantitative information was found on the effects of moisture and weathering. Accordingly, the data presented have reference primarily to new roofs.

B. Comments on the Data

The materials listed are in part of mineral origin and in part of organic origin. Magnitudes of their unit heat capacities and, to a lesser extent, their densities reflect their origin. The thermal conductivities show magnitudes which further demonstrate the physical nature of the different materials. The

TABLE 7—THERMO-PHYSICAL PROPERTIES OF ROOFING MATERIALS[a]

MATERIAL	REFERENCE[b]	c_p UNIT HEAT CAPACITY Btu/(Lb)(F Deg)				ρ DENSITY (Lb)/(Cu Ft) ORDINARY TEMPS		k THERMAL CONDUCTIVITY Btu / (Hr)(Sq Ft)(F Deg/Ft)		$k\rho c_p$ $\frac{(Btu)^2}{(Hr)(Ft)^4(F\ Deg)^2}$	THERMAL RESISTANCE PER INCH THICKNESS $\frac{1}{12k}$	$a = \frac{k}{\rho c_p}$ THERMAL DIFFUSIVITY (Sq Ft)/Hr
		True	At Temp F	Mean	Over Temp Range F	Variation Range	Avg	k	Mean Temp F			
Rosin-sized Paper	15			0.33			70.	0.08		1.85	1.0	0.0035
30-Lb[c] Rag Base Felt, Asphalt Saturated	28, 29			0.27			65.	0.046 0.047	50 100	0.83	1.8	0.0027
55-Lb Asbestos Base Felt, Asphalt Saturated	28, 29			0.20			110.	0.1[d]		2.2	0.8	0.0046
15-Lb Rag Finishing Felt, Tar Saturated	28, 29			0.35			69.	0.046 0.047	50 100	1.1	1.8	0.0019
Asphalt Saturated				0.27			63.	0.046 0.047	50 100	0.80	1.8	0.0027
15-Lb Asbestos Finishing Felt, Asphalt Saturated	28, 29			0.20			73.	0.1[d]		1.5	0.8	0.0068
Moppings for Deck, Ply and Top Coatings	10, 12, 15, 27											
Coal Tar Pitch				0.35			81.	0.6[d]		17.	0.17	0.021
Asphalt				0.22			64.	0.43	68	6.6	0.19	0.031
Asphalt Cut-Back Residue				0.22			64.	0.43	68	6.6	0.19	0.031
Surfacing Slag (Avg 3 Lb per Sq Ft Roof)	15, 21			0.2			80.	0.064		1.0		0.0040
Surfacing Gravel (Avg 4 Lb per Sq Ft Roof)	21			0.2			100.	1.0–2.5		35.		0.088
Asbestos Cement Shingle	29			0.2			110.	0.25 0.26	50 100	5.6	0.33	0.012
Asphalt Slate-Dust Shingle	29			0.2			73.	0.084 0.091	50 100	1.3	0.92–0.99	0.0060
Slate, Across Grain	11, 17, 23			0.18		169–175	172.	0.74–0.87		25.	0.113–0.095	0.026
Built-up Roofing, Slag or Gravel Surface	11							0.11			0.76	
Tar Roofing	13						55.	0.059			1.4	

[a] Tabulated Magnitudes of ($k\rho c_p$), $1/12k$ and a are based upon the following values:
k as listed, mean where data given for two temperatures.
ρ average listed.
c_p mean listed.
No adjustment has been made to bring values of k, ρ, and c_p to the same temperature.

[b] See references on pages 34–35.

[c] Pounds per 100 sq ft.

[d] Estimated.

felts have some insulating effect, while the moppings are relatively good heat conductors. The average value of 0.11 for the thermal conductivity of built-up roofings, as given in THE GUIDE 1947, is seen to be a fair value. More information on the effects of temperature, moisture, and weathering would seem desirable.

It should be noted that the pitch and asphalt used for moppings may have varying properties according to their source and method of preparation. The available data are inadequate to establish the possible range of variation and should be taken as rough estimates only. The thermal conductivities for asbestos asphalt-saturated felt and coal tar pitch represent approximated data (with the aid of Fig. 8); experimental data are needed.

XII. EMPIRICAL EQUATIONS FOR ROUGH ESTIMATES OF THERMAL CONDUCTIVITY, THE PRODUCT ($k\rho c_p$), AND THERMAL DIFFUSIVITY

The fact that the thermo-physical properties of nonmetallic solids may be classified by the origin and density of the material was pointed out by Mackey and Wright [2] who proposed empirical equations for use in rough estimates where data were otherwise lacking. Their suggested equations for dry materials were:

Apparent Density, ρ, between 5 and 20 lb per cubic foot:

Vegetable or mineral origin:

$k = 0.0137\rho^{0.3}$ Btu per (hour) (square foot) (Fahrenheit degrees per foot)

Vegetable origin:

$k\rho c_p = 0.0045\rho^{1.3}$ (Btu)^2 per (hour) (foot)^4 (Fahrenheit degrees)^2

$a = 0.0417/\rho^{0.7}$ (square feet) per (hour)

Mineral origin:

$k\rho c_p = 0.0027\rho^{1.3}$ (Btu)^2 per (hour) (foot)^4 (Fahrenheit degrees)^2

$a = 0.0696/\rho^{0.7}$ (square feet) per (hour)

Apparent Density, ρ, between 20 and 60 lb per cubic foot:

Vegetable or mineral origin:

$k = 0.00118\rho^{1.12}$ (Btu) per (hour) (square foot) (Fahrenheit degrees per foot)

Vegetable origin:

$k\rho c_p = 0.00039\rho^{2.12}$ (Btu)^2 per (hour) (foot)^4 (Fahrenheit degrees)^2

$a = 0.00357\rho^{0.12}$ (square feet) per (hour)

Mineral origin:

$k\rho c_p = 0.00024\rho^{2.12}$ (Btu)^2 per (hour) (foot)^4 (Fahrenheit degrees)^2

$a = 0.0058\rho^{0.12}$ (square feet) per (hour)

Apparent Density, ρ, of 60 lb per cubic foot and higher:

Either origin:

$k = (\rho/143)^{2.5}$ (Btu) per (hour) (square foot) (Fahrenheit degrees per foot)

Either origin:

$k\rho c_p = (\rho/55)^{3.5}$ (Btu)^2 per (hour) (foot)^4 (Fahrenheit degrees)^2

Either origin:

$a = (\rho/1330)^{1.5}$ (square feet) per (hour)

The apparent density noted is based upon the external dimensions of a sample of the material; as such, it includes the influence of air spaces in naturally porous or loose-packed material.

In Fig. 8 the preceding equations for thermal conductivity, k, are compared

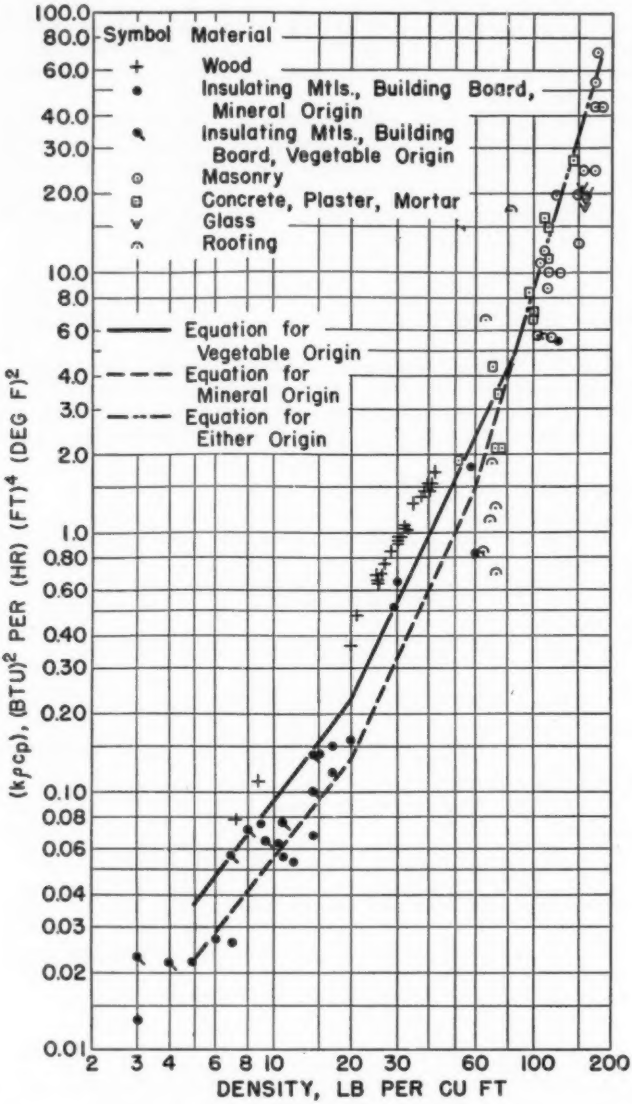


FIG. 9. COMPARISON BETWEEN DATA OF THIS REPORT AND EMPIRICAL ESTIMATING EQUATIONS FOR $(k\rho c_p)$

with the data given in this Bulletin. Fig. 9 presents the companion comparison for the product, $(k\rho c_p)$, and Fig. 10 for the thermal diffusivity, a.

The empirical equations evidently are adequate for their intended rough-estimating purpose. Several points in the figures are 50 percent or more off the empirical curves, but even such a magnitude of uncertainty is preferable to no information at all. Since any materials which have not been listed in this Bulletin may, at best, be judged similar to some of those which have been

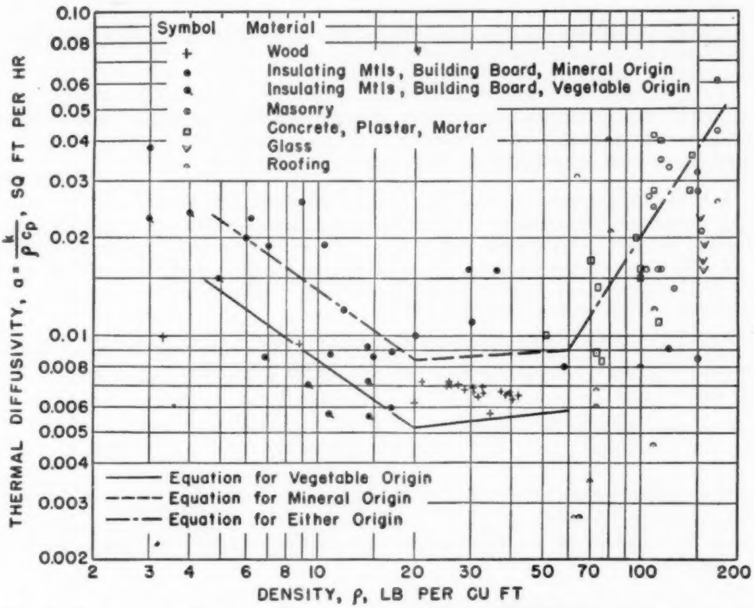


FIG. 10. COMPARISON BETWEEN DATA OF THIS BULLETIN AND EMPIRICAL ESTIMATING EQUATIONS FOR THERMAL DIFFUSIVITY

included, Figs. 8, 9, and 10 should serve as convenient aids in estimating thermo-physical properties of various materials.

XIII. DISCUSSION OF THE DATA PRESENTED

From the investigation conducted in preparing this Bulletin, it is concluded that:

1. Sufficient data on thermo-physical properties have been compiled in this Bulletin to meet the ordinary practical requirements of air conditioning load estimates, including periodic heat flow through walls and roofs.

2. Examination of the available data on thermo-physical properties has disclosed a lack of thorough and systematic information for many important building materials,

particularly regarding the effects of variable composition, moisture content, and temperature.

While the present data on thermo-physical properties can be used, they should be regarded, collectively, as a compromise compilation. The situation may be summarized as follows:

1. Data for wood and glass are in good order.

2. Data for building and insulating boards and insulating materials are in fair order.

3. Data on masonry materials, concrete, plaster and mortar, and roofing materials could well be extended and improved.

To assist the development and design engineer, the research worker, and others interested in steady state and periodic heat flow, it would be desirable to have data available on the thermo-physical properties of building materials established under the following conditions:

1. Obtained by standardized or approved experimental techniques.

2. Tested over the entire practical range of composition, temperature, and moisture content, the results being accompanied by a statement of the accuracy represented.

3. The range of test conditions established by a central agency, and all data approved by this agency prior to general publication. Such data would include comprehensive details of the materials tested.

The present study is a preliminary survey and cannot be expected to conform to the ideal requirements given; its intention is to portray the existing state of affairs.

ACKNOWLEDGMENTS

The author gratefully acknowledges the cooperation of individuals and organizations contributing to the preparation of this Bulletin. He is particularly appreciative of the unpublished data and reports received from the following:

1. *Structural Clay Products Institute.*
2. *Portland Cement Association.*
3. *National Bureau of Standards.*
4. W. Stanley Miles, Johns-Manville Corporation.
5. R. H. Heilman, Mellon Institute of Industrial Research.
6. V. Sanders, Pittsburgh Corning Corporation.
7. J. C. Hostetter, Mississippi Glass Co.

REFERENCES

[1] A.S.H.V.E. RESEARCH REPORT No. 1255—Periodic Heat Flow—Homogeneous Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 293-312).

[2] A.S.H.V.E. RESEARCH REPORT—Periodic Heat Flow—Composite Walls or Roofs, by C. O. Mackey and L. T. Wright, Jr. (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping, and Air Conditioning,* June 1946, p. 107-110).

[3] Effect of Moisture Changes on the Shrinking, Swelling, Specific Gravity, Air or Void Space, Weight and Similar Properties of Wood, by J. D. MacLean (*U. S. Forest Products Laboratory* Report R1448, August 1944).

[4] The Specific Heat of Wood, by F. Dunlap (*U. S. Department of Agriculture* Forest Service Bulletin, No. 110, 1912).

[5] Thermal Conductivity of Wood, by J. D. MacLean (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 323-354).

[6] Thermal Conductivity, Expansion and Specific Heat of Insulators at Extremely Low Temperatures, by G. B. Wilkes (*Refrigerating Engineering*, July 1946, p. 37-42, 68-72).

[7] The Specific Heat of Thermal Insulating Materials, by G. B. Wilkes and C. O. Wood (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 493-504).

[8] The Properties of Glass, by G. W. Morey (Reinhold Publishing Corp., New York, 1938).

[9] Handbook of the Glass Industry, by S. R. Scholes (Ogden-Watney Publishers, Inc., New York, 1941).

[10] Industrial Heat Transfer, by A. Schack (John Wiley & Sons, New York, 1933, p. 343-349).

[11] HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1947, Chapter 6.

[12] Chemical Engineers' Handbook, by J. H. Perry (McGraw-Hill Book Co., New York, 1934, 1st Edition, p. 485).

[13] Loc. cit., ref. 12, p. 2189.

[14] Loc. cit., ref. 12, p. 397.

[15] Loc. cit., ref. 12, p. 827.

[16] International Critical Tables (McGraw-Hill Book Co., New York, 1928, Vol. 2, p. 312).

[17] Loc. cit., ref. 16, Vol. 2, p. 52-55.

[18] Loc. cit., ref. 16, Vol. 2, p. 119.

[19] Thermal Conductivity of Insulating Materials at Low Mean Temperatures, by F. B. Rowley, R. C. Jordan, and R. M. Lander (*Refrigerating Engineering*, December 1945, p. 541).

[20] Heat Transmission, by W. H. McAdams (McGraw-Hill Book Co., New York, 1942, 2nd edition, p. 382).

[21] Mechanical Engineers' Handbook, by L. S. Marks (McGraw-Hill Book Co., New York, 4th edition, 1941, p. 548).

[22] Loc. cit., ref. 21, p. 393.

[23] Die Wärmeübertragung, by M. ten Bosch (J. Springer: Berlin, 1936, 3rd edition, p. 246-251).

[24] Low Mean Temperature Thermal Conductivity Studies, by F. B. Rowley, R. C. Jordan, and R. M. Lander (*Refrigerating Engineering*, January 1947, p. 35-39).

[25] Private Communication, *Structural Clay Products Institute*, Washington, D. C.

[26] Private Communication, *Portland Cement Association*, Chicago, Ill.

[27] Private Communication, *National Bureau of Standards*, Washington, D. C.

[28] Private Communication, by W. Stanley Miles (Johns-Manville Corp., New York, N. Y.).

[29] Private Communication, by R. H. Heilman (Mellon Institute of Industrial Research, Pittsburgh, Pa.).

[30] Private Communication, by V. Sanders (Pittsburgh Corning Corp., Pittsburgh, Pa.).

[31] Private Communication, by J. C. Hostetter (Mississippi Glass Co., St. Louis, Mo.).

1316

No. 1316

FORCED CONVECTION HEAT TRANSFER FROM FLAT SURFACES

Part I—Smooth Surfaces

By George V. Parmelee* and Richard G. Huebscher,** Cleveland, Ohio

This paper† is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio

I. Introduction

Nature and Scope of the Investigation

THE transfer of heat by forced convection at the exterior surfaces of buildings is a factor in calculations leading to the design of heating and cooling systems. In estimating steady state heat flow through walls of relatively low conductance, the convection coefficient is usually of secondary importance. It is of primary importance, however, in treating periodic heat flow as it is influenced by weather elements, especially solar radiation. It is also of great importance in calculating heat flow through building elements, such as glass, which have a relatively high thermal conductance. Outside the field of heating and air conditioning there can be cited many examples of technical problems in which convection heat transfer from flat surfaces is involved.

This bulletin reports the results of experimental measurements of the rate of heat transfer by forced convection from a vertical smooth flat surface swept by a horizontal parallel air stream. Data are given for turbulent, laminar, and transitional boundary layer flow along the surface and show the combined and separate effects of air velocity and surface length. The data are compared with skin friction theory through the medium of dimensionless numbers. Data of similar nature found in the literature are also cited. A report [1] on turbulent flow heat transfer was published in 1946.

II. Fluid Flow Principles

Flow of Fluids in Pipes

It is common knowledge that the velocity of a fluid flowing in a pipe is not constant at all points in any given cross section. At the surface of the pipe the fluid velocity is zero. This condition is confined to an extremely thin fluid layer, so thin that measurements within it can be made only with great difficulty. However, measurements have been made close enough to the pipe wall to substantiate this conclusion. As the distance from the pipe wall increases, it is found that the velocity of the fluid increases, reaching its maximum at the

* Research Fellow, A.S.H.V.E., Research Laboratory. Member of A.S.H.V.E.
** Assistant Research Engineer, A.S.H.V.E. Research Laboratory. Associate Member of A.S.H.V.E.
† Published as A.S.H.V.E. Research Bulletin, Vol. 53, No. 3.
[1] Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Coronado, Calif., June 1947.

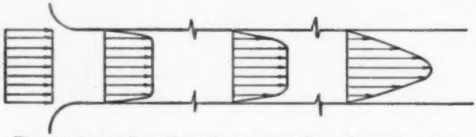


FIG. 1. VELOCITY DISTRIBUTION IN A DUCT

center. This point-to-point variation in velocity along the pipe diameter is influenced by the viscosity of the fluid. The actual velocity distribution is dependent upon the inertia of the fluid mass and the shearing or viscous action (skin friction) between the fluid and the pipe wall. The ratio of these two forces is the Reynolds number, N_{Re}.

It is not always recognized that the velocity profile of the fluid changes as the fluid moves along the axis of the pipe. Near the entrance to the pipe (assuming, for example, a pipe withdrawing fluid from a tank at a constant rate) the velocity is substantially constant across the pipe cross section except very close to the wall. Farther downstream a greater portion of the fluid is retarded by the viscous drag at the pipe wall and it will be found that the

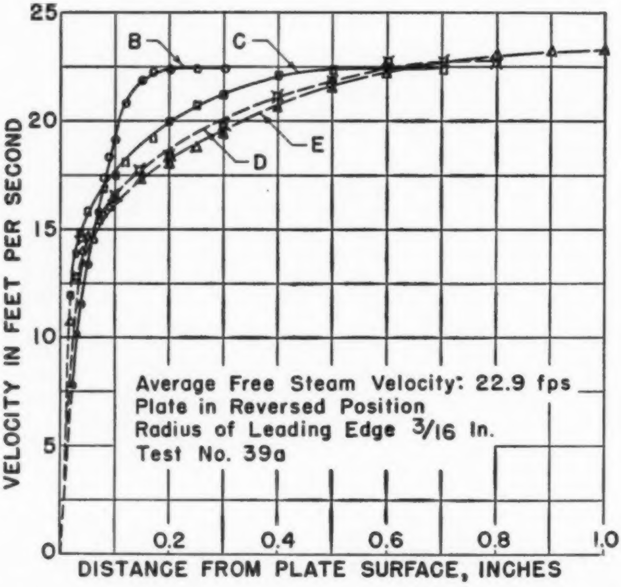


FIG. 2. STUDY OF VELOCITY DISTRIBUTION IN BOUNDARY LAYER ALONG THE SMOOTH FLAT PLATE

velocity at the center of the pipe exceeds the center velocity of the entrance section. The velocity profile undergoes a continuous modification with respect to position along the pipe axis until, at a sufficiently great distance from the entrance, the effect of viscosity reaches to the central portions of the fluid stream. No further changes are possible without a change in the fluid properties themselves. Obviously, the smaller the pipe the closer to the entrance is this final form reached. This discussion is illustrated by Fig. 1. The exact form of the profile is some function of N_{Re}, in which the characteristic dimension is the pipe diameter.

Flow of Unconfined Fluids Along Smooth Surfaces

Along a surface (such as a building wall) swept by a parallel *unconfined* air stream, the velocity profile of the stream next to the surface undergoes a

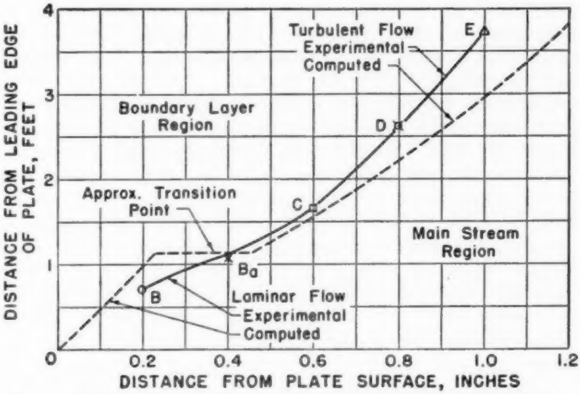


FIG. 3. BOUNDARY LAYER CONTOUR ALONG THE FLAT PLATE

continuous modification. The characteristic dimension affecting the value of the Reynolds number is the length of surface passed over by the air stream. Fig. 2 shows velocity profiles measured at various points along the plate during the course of one of the tests described in this bulletin. It is important to note that there is nothing in these velocity studies, in similar studies made by others, or in the theoretical treatment of the subject to indicate that the velocity profile for the parallel flow of air along a smooth surface will not be continuously modified no matter how long the surface. The transformations in the velocity profile are most severe near the leading edge and become less marked as the distance from the leading edge of the surface increases. It makes no difference, of course, whether the surface moves, be it railroad car or an airplane wing, or whether the air moves.

Those layers of air which undergo a change in velocity constitute what is known as the boundary layer. Fig. 3 shows the dimensions of the boundary layer determined from the data which are plotted in Fig. 2. Its thickness was taken to be that distance from the plate beyond which no change in velocity

could be measured. Also shown in Fig. 3 is the computed boundary layer contour. In fully developed pipe flow, the boundary layer thickness is equal to one-half the pipe diameter. The behavior of the fluid within this layer is quite complex. There may be three distinct patterns comprised of a laminar flow in a layer next to the plate, turbulent flow in an outer zone, and a transition flow in the intermediate transition or buffer zone. The boundary layer flow is termed turbulent if these three patterns are present. The velocity distribution at any point is a power function of the ratio of the distance from the surface to the boundary layer thickness at that point. Although the exponent is commonly taken as 1/7, recent investigations have shown that this exponent decreases as the Reynolds number increases.

In the case where only the laminar flow pattern exists, the boundary layer is termed laminar. In pipe flow the velocity distribution for this flow can be shown to follow a parabolic curve. In analytical studies of flow past flat surfaces it is usually assumed that this distribution also obtains. Theoretical equations based upon the parabolic velocity distribution confirm this assumption by their agreement with skin friction test data. However, velocity explorations within the laminar portion of a turbulent boundary layer show a linear velocity distribution.

In pipe flow, when the Reynolds number exceeds a limiting value, the flow changes from laminar to turbulent. Likewise, in flow along a flat surface, the boundary layer flow changes from the laminar to the turbulent form. This transition is affected by surface conditions, large scale eddying of the air stream, and other factors. Under some circumstances the flow over the forward portion of the surface may be laminar and that over the remainder turbulent, or the flow may be entirely laminar or entirely turbulent.

Obviously the character of the fluid flow over a surface will be closely related to the rate of convective heat transfer. It is also apparent that the unit rate of heat transfer will vary from point to point along the surface. In the analogous phenomenon of skin friction, the friction force likewise varies from point to point.

III. TEST APPARATUS

Specifications of the Test System

In order to set up a test system that duplicates at least in part the conditions that exist in practice, consideration is given first to the manner in which air flows around and over a building, since the primary concern here is with the transfer of heat to or from building surfaces. Let it be assumed that the air stream flows over a flat level expanse of ground after which it strikes perpendicular to the front wall of a single isolated building. Obviously the flow pattern is considerably modified. The front wall surface becomes a stagnation area, because the velocity of the wind perpendicular to the surface is reduced to zero. A corresponding increase in static pressure results, and the flow becomes roughly parallel to the surface, moving upward toward the roof and outward toward the sides, where it becomes more or less parallel to these surfaces. At the rear wall, in the lee of the wind, there is relatively little air motion. The design of the building and its position with respect to others have much to do with the flow conditions which exist at various points along

its outer surface. The following are some general considerations affecting flow conditions:

(a) The property of fluid viscosity causes wall surfaces to exert a retarding action on a limited portion of the air stream.

(b) The air stream in its approach to the building may flow in parallel straight lines and be at uniform velocity at all points, or there may be a superimposed large scale turbulence consisting of eddying currents caused by obstructions, abrupt changes in direction, and the like.

(c) The air stream is completely unconfined or is sufficiently large so that the reaction between it and the wall surface is unaffected by similar reactions at other points in the stream.

(d) The building surfaces over which the air passes may vary greatly in roughness. The surface may be smooth, or of uniform roughness, such as concrete or brick. It may be broken up by recessed or projecting areas, or it may be broken up by rib or finlike projections located according to some system or at random.

It can be seen that more variables are involved than can be evaluated in a single laboratory set-up. Study of a simple case must precede study of the more complex cases. The simplest case is that of flow parallel to a smooth surface. The test apparatus, which is described in detail later, was designed to duplicate certain features of the characteristics of practical flow conditions described. The specific characteristics of the test system were as follows:

(a) The surface tested was smooth and flat except for the rounded edge facing the air stream.

(b) The air stream flowed in parallel straight lines and was of uniform velocity at all points except near the surfaces of the wind tunnel which was used to produce the flow.

(c) Large scale turbulence was small and was accurately defined.

(d) The air stream was confined but the heat transfer surface was located in such a way that the reaction between the confining tunnel and the air stream in no way affected the reaction between the test surface and the air stream.

Construction of the Plate

The heat transfer element was an electrically heated composite structure, of which the outer surfaces consisted of polished aluminum plates. The whole was suspended from streamline struts in the center of a draw-through type wind tunnel. The principal dimensions of both the plate and the tunnel are shown in Fig. 4, which also gives the details of the strut section that formed the leading and trailing edges of the plate.

The heating section consisted of four flat heater elements wound on asbestos board with nichrome ribbon and laid end-to-end ¼ in. apart. Surrounding these, separated by a ¼ in. gap and lying in the same plane, was a hollow rectangle of asbestos cement board on which were wound four separate guard windings as shown in Fig. 4. This heating section was sandwiched between two ¼-in. sheets of asbestos board in which saw slots outlined each of the four heater plates and their respective guard sections. Several sheets of soft asbestos paper were placed next. Finally the whole was sandwiched between the sheets of ⅛-in. polished aluminum and fastened together with countersunk-

head brass screws. Fig. 5 shows an enlarged view of this assembly. A wiring diagram for the plate and guard area heaters and the control circuit is given in Fig. 6.

Surface Temperature Measurement

The surface temperature of the aluminum plate was measured by copper-constantan thermocouples set in grooves in the underside of the plate. The junctions and wire leads were electrically insulated by a thin coating of Sauereisen cement. Five junctions were located on each side opposite each

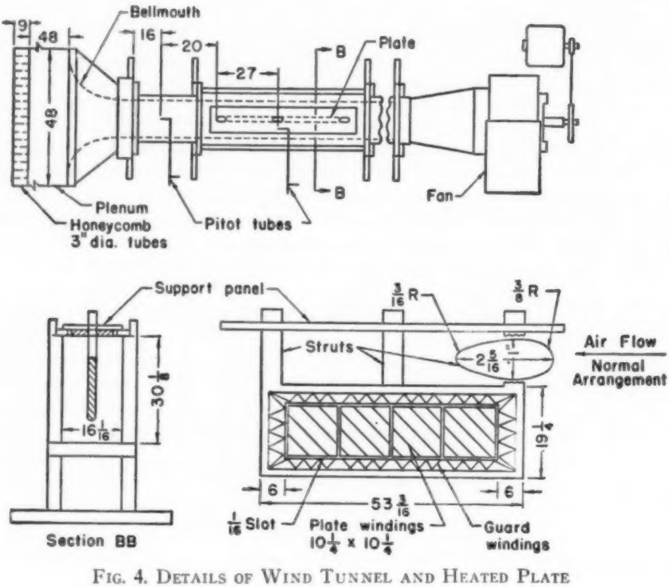


FIG. 4. DETAILS OF WIND TUNNEL AND HEATED PLATE

square heater plate, four of which were at the centers of equal areas and the fifth at the center of the plate. Thermocouples were also located opposite the center of each guard winding. In all there were 60 such couples. All wires including the leads to the heater windings were run in rubber tubes carried up through the struts to a terminal panel located on the support panel. The constantan lead wires terminated in cold junctions set in insulated ice boxes which also were located on the plate support panel. Copper leads connected the terminals to multiple-point switches.

Control of Guard Surface Temperature

To indicate edgewise heat flow from the test areas differential couples were cemented to the ¼-in. asbestos board with junctions located on each side of the saw slots. The input to the guard winding was then adjusted until these

couples showed a zero emf. Differential couples were also located on sides of the slots between the test areas to permit testing a single area, which was then guarded by the adjacent heater plates as well as by the appropriate guard winding. These guard couples consisted of groups of two, four, and six couples connected in series and located so that the temperature differential across different boundaries between the guard and guarded areas could be checked during the balancing process. In testing, when satisfactory balance was achieved, the proper groups were connected in series leaving two leads that were connected to a sensitive mirror type galvanometer. When temperature balance existed (see Fig. 6), a light beam reflected from the mirror struck a photocell unit and caused it to hold open the contacts of a relay. When there

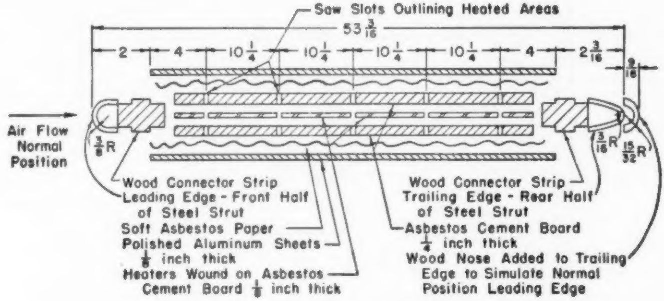


FIG. 5. ENLARGED SECTION OF TEST PLATE

was a temperature unbalance, the beam was diverted from the photocell, and the relay supplied additional heat to the guard windings.

Air Flow System

The principal dimensions of the draw-through type tunnel are shown in Fig. 4. An external view of the assembled test equipment as obtained during a test is shown in Fig. 7. The plate assembly with the cold junction ice boxes, part of wind tunnel, and photocell temperature control system are shown in Fig. 8.

The tunnel was constructed of 5/16-in. hard board and painted flat black inside. Straightening tubes of 4-in. stove pipe 24 in. long were placed in the exit end of the tunnel to eliminate any possible swirl induced by the fan entrance. Regulation of the rate of air flow was obtained in three ways: (1) by change of fan speed, (2) by restriction of the fan outlet, (3) by use of bleeder openings near the fan inlet permitting air to bypass the tunnel.

The air velocity was measured by ¼-in. diameter Pitot-static tubes held in traversing brackets. These brackets in turn were bolted to a horizontal rail running lengthwise of the tunnel. This rail could be moved up or down so that the velocity could be explored at any point. Most velocity pressure measurements were made with calibrated inclined manometers. At very low velocities

a hook gage easily readable to 0.001 in. water was used. It was read four times by each of two observers during the tests. The maximum deviation from the mean reading did not exceed 0.0005 in. of water, which represented, at the lowest velocity of 3 fps, a possible error in velocity of about 10 percent.

The fan used in these tests produced a flow of about 10,000 cfm at 1260 rpm and velocities of 37 mph just in front of the test plate and 40 mph in the channel between the plate and the tunnel wall.

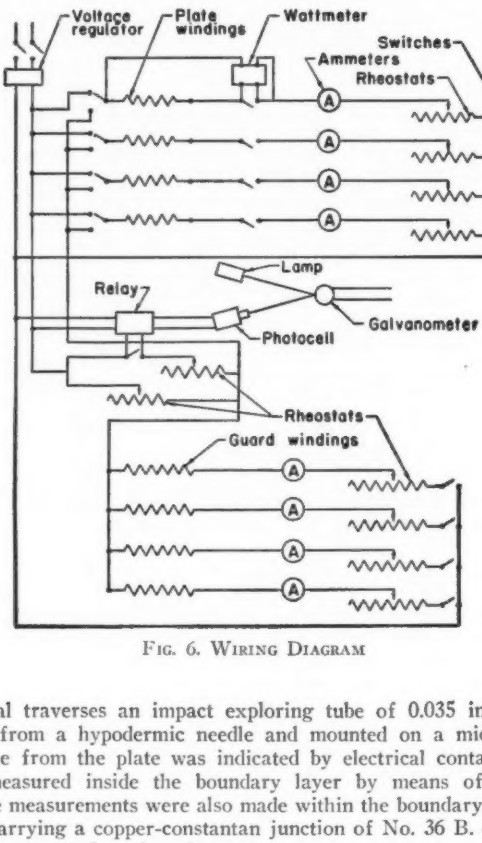


FIG. 6. WIRING DIAGRAM

For special traverses an impact exploring tube of 0.035 in. diameter was constructed from a hypodermic needle and mounted on a micrometer screw. Zero distance from the plate was indicated by electrical contact. Static pressure was measured inside the boundary layer by means of a static tube. Temperature measurements were also made within the boundary layer by means of a probe carrying a copper-constantan junction of No. 36 B. & S. gage wire. This also was mounted on the micrometer screw.

Evaluation of Turbulence or Large Scale Eddying of Air Streams

Considerable attention was given to insuring uniform flow conditions in the tunnel. It was believed that this could be done by a properly designed bell-

mouth entrance (shown in Fig. 4). This was proportioned in accordance with the *A.S.M.E.* recommendations for flow nozzles, slightly modified to suit the rectangular shape of the tunnel. A 5-in. wide flange in the plane of the entrance was also incorporated. Following completion of the tunnel and entrance section a study was made of the flow pattern in the tunnel. Two criteria were used to judge the flow conditions. One was a comparison of the velocity profiles at different levels of various sections of the tunnel by Pitot tube traverses. The second was an evaluation of the large scale turbulence, or eddying, of the tunnel flow by determining the critical Reynolds number at which there takes place an abrupt drop in the ratio of pressure drop across a sphere to the fluid velocity pressure. This is a standard method of measuring the turbulence in aerodynamic wind tunnels and is a result of a study [2] undertaken to discover why, when identical air foils were tested in different tunnels, different drag coefficients were measured. Turbulence is defined as "the ratio of the root-mean-square value of the deviations of the air speed from its mean value to the mean value." A turbulence of one percent means an equivalent maximum wind fluctuation of plus or minus 1.4 per cent from the mean value. This turbulence consists of large scale eddying flow that exists in a stream of air moving at what appears to be a uniform and constant velocity and is not to be confused with eddying flow influenced by viscous drag between a surface (such as a pipe wall) and a fluid wherein the velocity distribution may be described as turbulent.

The drag coefficient of a sphere, and the more readily measured pressure drop, are very sensitive to eddying or large scale turbulence of the main stream, because such turbulence affects the point of separation of the boundary layer relative to a diameter through the sphere normal to the flow. A consequence of this separation is the wake, a region of reduced pressure, in the rear of the sphere. The area of this wake is chiefly responsible for the drag and at sufficiently high values of the Reynolds number is practically constant in cross section. Therefore the drag coefficient becomes practically constant. The critical transition value of the ratio of pressure drop to velocity pressure is about 1.22 (corresponding to a drag coefficient of 0.30).

The percent turbulence is a function of the critical Reynolds number as follows [3] (the sphere diameter is used in evaluating the Reynolds number):

PERCENT TURBULENCE	CRITICAL N_{Re}
0.0	385,000 approx.
0.5	270,000
1.0	232,000
1.2	197,000
1.6	164,000
2.3	132,000

Turbulence Measurements in the Wind Tunnel

Turbulence in the wind tunnel was evaluated by means of a 7-in. diameter sphere. Turbulence in the original installation with no baffles or plenum chamber was, as shown in Fig. 9, of the order of 3.5 percent. The addition of the plenum chamber and honeycomb (followed by window screen) reduced the turbulence to about 1.5 percent at the front of the tunnel. At the position

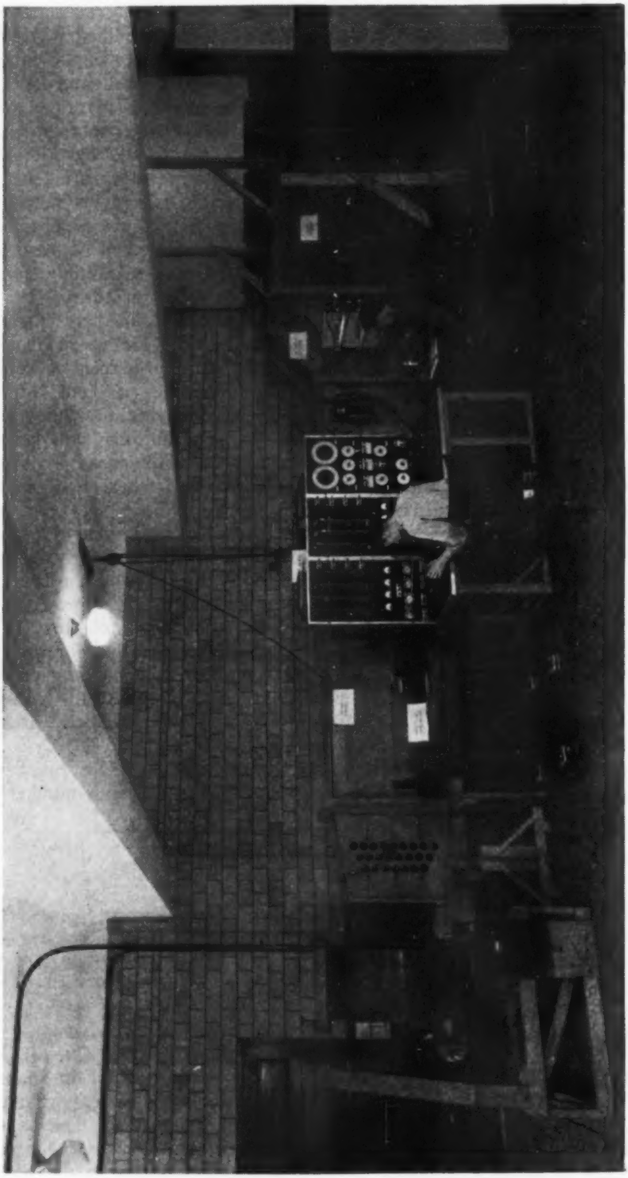


FIG. 7. WIND TUNNEL AND TEST EQUIPMENT ASSEMBLY

occupied by the test plate the turbulence was about 1.3 percent. Because of the confined quarters it was also necessary to resort to considerable baffling near the entrance to the plenum chamber in order to obtain flow symmetry of the return air from the fan. The velocity distribution was measured with

FIG. 8. PLATE ASSEMBLY WITH THE COLD JUNCTION ICE BOXES, PART OF WIND TUNNEL AND PHOTOCELL TEMPERATURE CONTROL SYSTEM

a ¼-in. diameter Pitot-static tube at four levels near the tunnel entrance (see Fig. 10), and between the test plate and the tunnel wall (see Fig. 11). The profiles are seen to be perfectly flat except close to the surfaces and are uniform from top to bottom within reasonably close limits. The velocity at the center of the channel between the plate and the tunnel is seen to be much greater at station B than at station A. This is due to the change in cross sectional area of the tunnel because of the space occupied by the test plate. From B to the rear the velocity increases slowly. As the fluid layers near the surface of the plate and the tunnel are retarded, the velocity of the main portion must of necessity increase in order that the total flow be constant. The curve dips more sharply at the right hand side of the figure because the

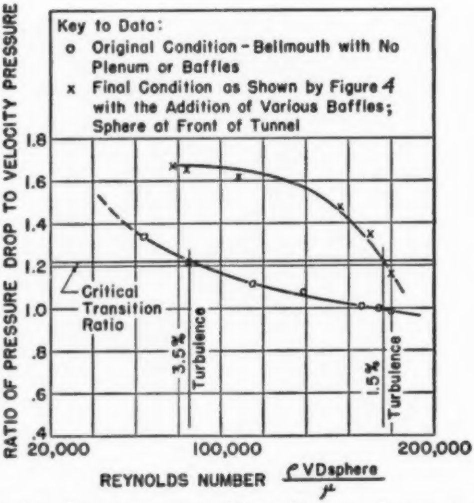


FIG. 9. TURBULENCE STUDIES OF THE WIND TUNNEL SHOWING PRESSURE DROP ACROSS 7-IN. DIAMETER SPHERE DIVIDED BY VELOCITY PRESSURE AS A FUNCTION OF REYNOLDS NUMBER

retarding effect of the tunnel wall has been exerted for a greater distance. The values at 7 in. from the plate correspond to ½ in. from the tunnel wall.

IV. TEST PROCEDURE

Leading Edge Conditions

Three arrangements of the plate itself were employed. In the *normal* arrangement the leading edge of the strut had a radius of ⅜ in. Later the plate was

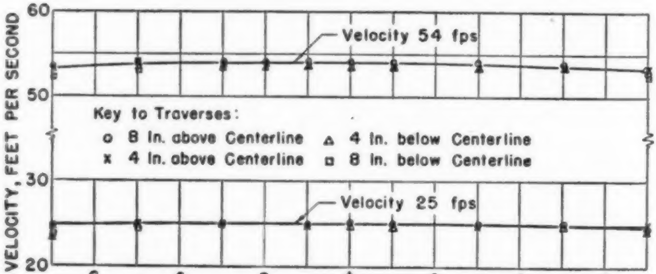


FIG. 10. VELOCITY DISTRIBUTION IN WIND TUNNEL WITH HONEYCOMB AND BAFFLES IN PLACE

tested in the *reversed* arrangement. Here the leading edge had a radius of 3/16 in. and was tapered more gradually. The plate was tested in the reversed arrangement with the leading edge *built up* to 15/32 in. in radius and faired into the plate with model airplane paper to simulate the abruptness of the nose in the normal arrangement. A few tests were made wherein the rear portion of the plate was heated and the forward portion unheated.

The apparatus was located in the lower part of the laboratory building in order to take advantage of the relative constancy of the air temperature. During the period of a test the temperature change was small enough so that heat storage effects were negligible.

Heat Input to Plate

Even with a pre-selection of velocity and effective length considerable choice was possible in the details of heating the test areas. By Method I, the input to each test area could be adjusted so that there was no heat transfer from one

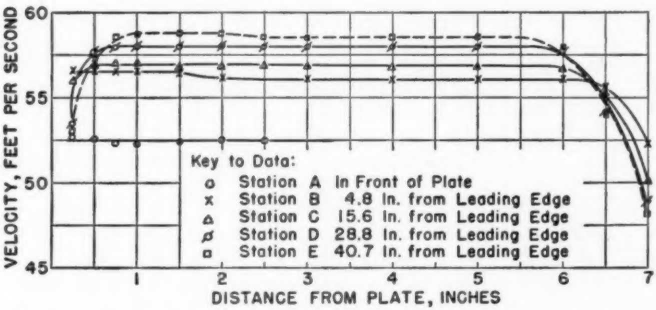


FIG. 11. VELOCITY DISTRIBUTION IN WIND TUNNEL WITH PLATE IN POSITION

area to the next. By Method II, equal inputs could be supplied each area and the individual areas treated as a group. Both methods were used and no great difference in the results was noticeable for otherwise identical conditions. The heat transfer from individual test areas could not, of course, be computed when Method II was followed. In both methods the heat input supplied was usually dictated by the air velocity. At very low velocities, for instance up to about 10 fps, the input was set at about 10 watts per test area to suppress natural convection tendencies due to temperature difference. At higher velocities the input was increased to 20, 40, and 60 watts so as to secure a plate-to-air temperature difference of not less than 10 deg.

Observations

The procedure in securing temperature equilibrium of the apparatus consisted of supplying heat to the plate for about a half hour after which the fan was put in motion. Temperature balance was then carried out by manual adjustment of the input to the various windings and could be secured in from 3 to 6 hr, whereupon the automatic system was given control. Once equilibrium

TABLE 1—SUMMARY OF TEST RESULTS
(Plate in Normal Position; Leading Edge Radius 3/8 In.)

h	L	V FPS	Temperature F Air	Plate	N_{Re}	$(N_{St})(N_{Pr})^{2/3}$	Y/L	α	Test No.
2.52	1.19	10.7	76.3	119.8	0.67×10^5	$3.24/10^3$	0.28	0.86	5
3.17	1.19	10.8	80.0	108.6	0.71	4.01	0.28	0.86	12
3.59	1.19	15.3	76.7	107.6	1.00	3.15	0.28	0.86	4
4.82	1.19	21.6	80.5	103.2	1.43	2.98	0.28	0.87	3
6.26	1.19	34.8	80.4	97.8	2.34	2.42	0.28	0.88	8
7.82	1.19	40.3	79.8	93.9	2.64	2.62	0.28	0.88	2
7.77	1.19	42.9	83.9	95.0	2.87	2.42	0.28	0.88	13
8.14	1.19	43.3	77.3	89.5	2.93	2.49	0.28	0.88	14
7.35	1.19	44.0	80.0	94.6	3.00	2.43	0.28	0.88	10
8.74	1.19	57.9	81.1	93.3	3.98	2.01	0.28	0.89	9
9.34	1.19	58.2	79.2	90.8	3.99	2.15	0.28	0.89	7
9.87	1.19	58.2	81.2	92.4	3.97	2.27	0.28	0.89	1
2.96	2.05	10.8	80.0	111.5	1.21	3.74	0.15	0.90	12
5.77	2.05	34.8	80.4	100.3	4.09	2.23	0.15	0.91	8
7.06	2.05	42.9	83.9	96.6	4.95	2.20	0.15	0.91	13
7.22	2.05	43.3	77.3	91.0	5.05	2.21	0.15	0.91	14
6.79	2.05	44.0	80.0	95.7	5.15	2.25	0.15	0.91	10
8.11	2.05	57.9	81.1	95.0	6.88	1.87	0.15	0.92	9
8.83	2.05	58.2	79.2	92.3	6.89	2.03	0.15	0.92	7
2.76	2.91	10.8	80.0	112.9	1.72	3.50	0.12	0.90	12
6.49	2.91	42.9	83.9	97.4	6.99	2.02	0.12	0.93	13
6.56	2.91	43.3	77.3	91.9	7.13	2.01	0.12	0.93	14
6.30	2.91	44.0	80.0	97.7	7.26	2.08	0.12	0.93	10
7.57	2.91	57.9	81.1	95.7	9.69	1.73	0.12	0.93	9
2.71	3.76	10.8	80.0	113.0	2.22	3.43	0.09	0.92	12
6.42	3.76	42.9	83.9	97.8	9.06	2.00	0.09	0.93	13
6.44	3.76	43.3	77.7	92.4	9.26	1.97	0.09	0.93	14
6.15	3.76	44.0	80.0	97.8	9.34	2.04	0.09	0.93	10
7.47	3.76	57.9	81.1	95.8	12.45	1.73	0.09	0.94	9

TABLE 2—SUMMARY OF TEST RESULTS
(Plate Reversed; Leading Edge Radius 15/32 In.)

h	L	V FPS	Temperature F Air	Plate	N_{Re}	$(N_{St})(N_{Pr})^{2/3}$	Y/L	α	Test No.
5.41	1.16	22.2	80.1	88.5	1.47×10^5	$3.24/10^3$	0.28	0.87	40
5.75	1.16	24.9	81.1	88.9	1.65	3.06	0.28	0.87	41
3.89	2.01	18.7	81.0	92.8	1.98	2.74	0.16	0.91	42
4.40	2.01	23.0	82.4	92.8	2.43	2.52	0.16	0.91	43
3.52	2.98	20.3	81.0	93.8	2.96	2.28	0.11	0.93	44a
3.35	3.72	20.3	81.0	94.1	3.76	2.17	0.09	0.93	44b
5.12	1.16	21.9	82.0	101.8	1.40	3.13	0.28	0.87	45
4.00	2.01	21.9	83.2	110.0	2.41	2.45	0.16	0.91	46
7.61	3.72	52.0	83.5	101.7	10.75	1.97	0.09	0.94	47

was established at one velocity, a new balance could be secured at a somewhat different velocity in 1 or 2 hr with only minor adjustments of the input to the heater windings.

Observations were as follows. Four sets of readings were taken over a period of 1 to 1½ hr. Indications of the couples located in the test areas were observed individually. In the guard areas couples on opposite sides were paralleled. Temperature measurements were accurate within ±0.5 F deg (calibrations of a number of samples from the same wire showed deviations from the mean curve not exceeding 0.1 F deg). The wet and dry bulb temperatures of the room air were observed together with the barometric pressure in order to estimate the specific humidity of the air in the wind tunnel where only the dry bulb temperature was observed. In order to make corrections for the radiant heat exchange between the plate and the tunnel, the temperature of each side of the tunnel wall was also measured. The precision potentiometer used in making all temperature measurements could indicate emfs to within 0.5 microvolts (0.025 F deg).

Except for the first five tests the electrical input to the test areas was measured by a precision wattmeter, accurate to one-quarter of 1 percent of the full scale deflection. At the input used, the error, except in several special tests, did not exceed 1 percent. The meter used in the first five tests was later carefully calibrated at the point of reading.

Computation of the Film Coefficient From Observed Data

A sample of computations for reducing test data to dimensionless quantities is given in Appendix A.

The unit rate of convective heat transfer was computed in the following manner. The electrical input, suitably corrected for scale factor and for power consumed by the meter itself, was converted into equivalent heat units. From this value was deducted the computed heat loss by radiation. The remainder, the net heat loss by convection, was then divided by the appropriate area and the difference between the plate surface and the air temperatures.

Surface temperatures were evaluated as follows. The average temperature of each test area was found by averaging the readings of the four couples located at the centers of equal areas of each test area, which were, of course, in pairs, one on each side of the plate assembly. The temperature corresponding to the average emf was then selected from the calibration curve. The average of the pair of values represented the temperature of the test area. The difference in temperature between opposite sides of the test surfaces rarely exceeded 1 deg.

The air temperature was computed from the average of three thermocouples located 6 in. in front of the plate's leading edge. Explorations in the channel between the plate and the tunnel wall showed that this temperature prevailed throughout and that the temperature rise of the air was confined to the boundary layer. Stratification of the air in the tunnel was rarely noticed. The temperature of the test surface was not constant throughout its length; that of the area nearest the leading edge was several degrees lower than at the rearward areas because the rate of heat transfer was greatest at this point.

The heat exchange by radiation between the test surface and the tunnel

TABLE 3—SUMMARY OF TEST RESULTS

(Mean Local Heat Transfer Coefficients of Four Separate Areas of the Plate; Plate in Normal Position; Leading Edge Radius 3/8 In.)

h	L	V FPS	N_{Re}	$(N_{St})(N_{Pr})^{2/3}$	Y/L	α	TEST No.
3.17	1.19	10.8	0.71×10^5	$4.01/10^3$	0.28	0.86	12
6.26	1.19	34.8	2.34	2.42	0.28	0.88	8
7.77	1.19	42.9	2.87	2.42	0.28	0.88	13
8.14	1.19	43.3	2.93	2.49	0.28	0.88	14
7.35	1.19	44.0	3.00	2.43	0.28	0.88	10
8.74	1.19	57.9	3.98	2.01	0.28	0.89	9
9.34	1.19	58.2	3.99	2.15	0.28	0.89	7
2.75	2.05	10.8	1.21	3.48	0.58	0.83	12
5.28	2.05	34.8	4.09	2.23	0.58	0.84	8
6.34	2.05	42.9	4.95	2.20	0.58	0.85	13
6.30	2.05	43.3	5.05	2.21	0.58	0.85	14
6.22	2.05	44.0	5.15	2.25	0.58	0.85	10
7.48	2.05	57.9	6.88	1.87	0.58	0.85	9
8.32	2.05	58.2	6.88	2.03	0.58	0.85	7
2.36	2.91	10.8	1.72	2.93	0.71	0.82	12
5.36	2.91	42.9	6.99	2.02	0.71	0.84	13
5.25	2.91	43.3	7.13	2.02	0.71	0.84	14
5.34	2.91	44.0	7.26	2.08	0.71	0.84	10
6.50	2.91	57.9	9.69	1.80	0.71	0.84	9
2.55	3.76	10.8	2.22	3.23	0.77	0.82	12
6.25	3.76	42.9	9.06	2.12	0.77	0.83	13
6.11	3.76	43.3	9.26	1.87	0.77	0.83	14
5.70	3.76	44.0	9.34	1.89	0.77	0.83	10
6.95	3.76	57.9	12.45	1.61	0.77	0.83	9

TABLE 4—SUMMARY OF TEST RESULTS

(Heat Transfer Coefficients of Heated Surfaces Preceded by Unheated Surface of Length X; Plate in Normal Position; Leading Edge Radius 3/8 In.)

h	L	X	V FPS	N_{Re}	$(N_{St})(N_{Pr})^{2/3}$	Y/L	α	TEST No.
6.94	1.71	2.22	43.8	4.27×10^5	$2.09/10^3$	0.50	0.85	11
4.64	2.57	1.36	30.7	4.47	2.01	0.34	0.88	16
6.34	2.57	1.36	43.1	6.34	1.87	0.34	0.88	15

wall was approximately 1 percent of the total heat flow. The emissivity of the polished aluminum was taken as 0.055.

The observed velocity pressures were corrected for gage calibration. The velocity was based upon the air density corresponding to the air temperature in the tunnel and the specific humidity of the room air. The velocities reported in the tables of data were those measured at station D, very nearly half way along the plate. The velocity at this point was close to the average of the several stations and was 10 percent greater than the velocity just in front of the plate. Fig. 2 shows typical velocity distribution at several points along the plate surface.

In computing dimensionless numbers the mean of the plate and air temperatures was used to evaluate the fluid properties. At the conditions of the tests the average air density was about 0.0700 lb per cubic foot.

When test Method I was used, not only the average rate of heat transfer for the entire length of surface but also the average rate of heat transfer for each of the four test areas could be computed, because they were guarded against edgewise heat flow. The data taken by Method II gave only the average coefficient for the surface tested. There was little difference in the average coefficient whether Method I or II was used.

V. EXPERIMENTAL DATA

Heat Transfer in Turbulent and Laminar Boundary Layers

Tables 1, 2, 3, and 4 contain the data obtained with a turbulent boundary layer flow along the surface under the conditions noted in the table headings. Each coefficient represents the average value for a surface made up of one, two, three, or four test areas each 10¼ in. long. The lengths indicated in the columns headed L equal one, two, three, and four times 10¼ in. plus the length of the heated guard surface between the nose or leading edge and the front edge of the first test area, (see Fig. 5). It was found that with the plate reversed, the boundary layer flow was considerably modified, remaining laminar up to N_{Re} of about 150,000. Therefore, after the data listed in Table 5 were obtained, the leading edge was built up to simulate approximately the leading edge conditions of the normal arrangement (Tables 1, 3, and 4). The data listed in Table 2 were taken with the plate reversed and with the leading edge built up.

Mean Local Heat Transfer Coefficients

Table 3 shows mean local heat transfer coefficients, that is, average coefficients for each of the four test areas. In these tests, from which part of the data in Table 1 was derived, each test area was guarded against edgewise flow not only along the line between the guard and the plate but also at the boundaries between adjacent test surfaces. The coefficients corresponding to $Y/L = 0.28$ are those for the first test area swept by the air stream, those corresponding to $Y/L = 0.58$ are for the second test area, and so on. For example for $Y/L = 0.28$, $Y = 4$ and $L = 4 + 10¼$; for $Y/L = 0.58$, $Y = 4 + 10¼$, $L = 4 + 10¼ + 10¼$ (see Fig. 5).

Heat Transfer from Heated Surface Preceded by Unheated Surface

Table 4 lists data for three tests in which the effect of an unheated surface preceding the heated test surface was investigated. In all tests reported in Tables 1, 2, 3, and 5 there were approximately 2 in. of unheated surface ahead of the guard surface and following test areas. This surface included the nose piece and a wood connector strip. However, in test 11, only the last test area was heated, and in tests 15 and 16, only the last two test areas were heated.

TABLE 5—SUMMARY OF TEST RESULTS

(*Plate Reversed; Leading Edge Radius 3/16 In.*)

h	L	V FPS	Temperature F		N_{Re}	$(N_{St})(N_{Pr})^{2/3}$ (Observed)[a]	$(N_{St})(N_{Pr})^{2/3}$ (Corrected)[b]	Test No.
			Air	Plate				
1.12	1.19	2.8	76.5	97.1	0.19×10^5	$5.25 / 10^3$	$4.88 / 10^3$	31
1.29	1.19	4.1	76.0	94.1	0.28	4.15	4.31	30
0.98	2.04	4.4	80.5	103.0	0.45	3.01	2.31	29
1.21	2.04	6.8	80.0	98.6	0.78	2.38	2.18	28
1.45	2.04	10.1	81.0	96.7	1.16	1.90	1.94	27
1.94	2.04	15.6	81.1	92.8	1.77	1.67	1.82	26
2.63	2.04	19.5	80.9	97.8	2.22	1.80	1.96	32
3.45	2.04	23.4	79.1	92.0	2.69	1.96	2.09	33
4.16	2.04	27.4	80.3	91.6	3.16	2.02	2.13	34
4.86	2.04	31.2	81.9	91.0	3.55	2.08	2.18	35
3.79	2.91	24.9	83.6	95.5	4.00	2.04	2.11	36
4.49	2.91	27.9	79.0	89.1	4.56	2.13	2.20	37
3.04	3.75	20.5	82.9	115.8	3.75	2.20	2.27	22
3.26	3.75	22.8	80.9	94.3	4.82	1.90	1.96	39b
3.64	3.75	26.2	79.7	107.8	4.90	2.05	2.12	20
3.87	3.75	26.4	78.1	104.2	4.95	2.18	2.24	21
4.11	3.75	27.8	83.7	94.5	5.82	1.98	2.04	38
4.49	3.75	30.9	82.2	107.1	6.21	1.97	2.03	19
6.24	3.75	43.2	81.5	99.7	9.28	1.92	1.95	18
6.24	3.75	43.3	79.5	96.5	9.12	1.91	1.94	23

[a] Based on experimental value of h.
[b] Based on h corrected for entrance length and natural convection.

In addition one test section was used as a guard section. Hence, this left various portions of the front part of the plate unheated.

VI. DISCUSSION OF RESULTS

Heat Transfer and Skin Friction Along Flat Surfaces

Osborne Reynolds early recognized that the phenomena of heat transfer and skin friction are analogous, the former being a diffusion of heat through fluid layers, the latter being a diffusion of momentum. Assuming the equality of heat and momentum diffusivity he proposed the relationship that N_{St}, the

Stanton number equals $\frac{1}{2}f$, the friction factor. This is approximately true for air. Numerous analytical studies have since been made to derive the proper relationship between momentum and thermal diffusivity. This relationship is expressed by the Prandtl number N_{Pr}, or by some complex function of N_{Pr}. Colburn[4, 5] has found that heat transfer and friction can be correlated quite simply through

$$(N_{St})(N_{Pr})^{2/3} = \tfrac{1}{2}f \quad (1)$$

The 2/3 exponent on N_{Pr} was derived from experimental results on heat transfer in pipes, which are usually expressed in the form of

$$(N_{Nu}) = f(N_{Re}, N_{Pr}) \quad (2)$$

The exponent on N_{Re} is 0.8. That on N_{Pr} varies from 0.3 to 0.4; Colburn used 1/3. Equation 2 can readily be transformed into the form of Equation 1 since f is a function of N_{Re}. In this instance the use of Equation 1 is particularly helpful because there are extensive data on skin friction of flat plates over a very wide range of Reynolds numbers, but only few data on heat transfer, and these only cover a limited range.

Goldstein[6] has compared the available friction data for flat smooth surfaces and for turbulent flow gives three equations for the friction factor as a function of N_{Re}. Satisfactory agreement with the data which he has correlated is given by the expression:

$$f = 0.455/(\log N_{Re})^{2.58} \quad (3)$$

Two other equations have been used to express f in terms of N_{Re}, as follows:

$$f = 0.074/N_{Re}^{0.2} \quad (4)$$

$$1/\sqrt{f} = 4.13 \log (N_{Re}f) \quad (5)$$

Equation 4 has been derived analytically on the basis of a 1/7 power velocity distribution for turbulent flow. It has the advantage of being simpler to use mathematically and, as has been shown by Colburn,[4, 5] fits the available heat transfer data rather well. The upper limit of the Reynolds number for the data used by Colburn was about 750,000. Goldstein[6] shows that most of the friction data for flat plates lie below this line for Reynolds numbers between 250,000 and 1,000,000 and considerably above it between 10,000,000 and 500,000,000. The discrepancy increases as N_{Re} increases. These data were obtained by tests in air on plates up to 9 ft in length, and, at the highest Reynolds numbers, by tests on plates in water, which has the advantage of a much lower kinematic viscosity (absolute viscosity divided by specific weight) than air. Equation 5 accurately represents the experimental friction data for turbulent flow but is awkward to use. Equation 3 is substantially the same as Equation 5 for values of N_{Re} between 200,000 and 10,000,000 and lies only 5 percent above it out to N_{Re} equals 500,000,000. Therefore, Equation 3 will be used in the heat transfer equation which, since the right hand member of Equation 1 equals $\frac{1}{2}f$, will be written:

$$(N_{St})(N_{Pr})^{2/3} = 0.2275/(\log N_{Re})^{2.58} \quad (6)$$

In support of this selection it should be stated that Wright,[7] in developing a new friction chart for round duct, pointed out that the Blasius expression

for pressure drop in pipes, on which the 1/7 power law for velocity distribution is based, is only approximate. He stated further that f vs N_{Re} plotted on logarithmic coordinates is not a straight line, according to the Blasius equation, but rather a curved line for both smooth and rough pipes, and further that for very large Reynolds numbers and for large roughness ratios, f becomes independent of N_{Re} and increases with the roughness ratio.

For the laminar boundary layer Goldstein[6] gives the skin friction equation as:

$$f = 1.327/N_{Re}^{0.5} \quad (7)$$

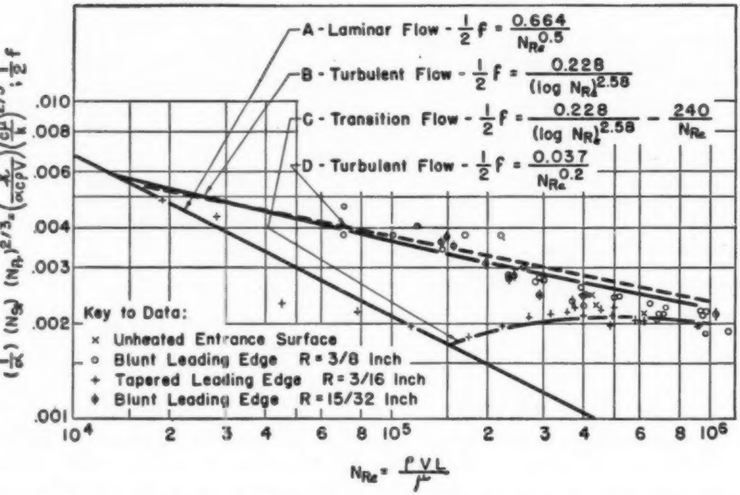


FIG. 12. CORRELATION OF HEAT TRANSFER COEFFICIENTS OF A FLAT PLATE WITH SKIN FRICTION FACTOR

This exact equation has also been analytically derived. The corresponding equation for heat transfer is:

$$(N_{St})(N_{Pr})^{2/3} = 0.664/N_{Re}^{0.5} \quad (8)$$

In connection with the discussion of friction it should be pointed out that there are two kinds of friction or drag. One, skin friction or drag, is a result of the shearing action between a surface and the fluid moving over it due to the fluid viscosity. The flow related to this can be quite accurately described mathematically. The second type of friction is variously referred to as form drag, profile drag, or eddy-making drag and is due to the wake, a region of reduced pressure and eddy currents in the rear of the body over which the fluid flows. For example, the drag of a flat plate placed *perpendicular* to the flow is largely eddy-making drag. When placed *parallel* to the flow the drag is largely skin friction drag. It should also be pointed out that while the correlation of heat transfer and friction for flow in pipes is very good, because it is

a matter of skin friction, the correlation of heat transfer for fluid flow over tube banks and pressure loss is very poor. This can be attributed to the fact that much of the pressure loss is due to eddy-making drag or friction. In the particular problem discussed here, good correlation can be expected. However, when eddy-making ribs or the like are added the correlation with friction will undoubtedly be poor.

Turbulent Boundary Layer Flow

The experimental data of Tables 1 and 2 have been plotted in Fig. 12 in terms of the dimensionless quantities N_{St}, N_{Pr}, and N_{Re}. The points shown include a fourth dimensionless factor α to account for the entrance conditions preceding the test area. The heat transfer measurements did not, of course, include the heat transferred from the guard surface adjacent to the nose

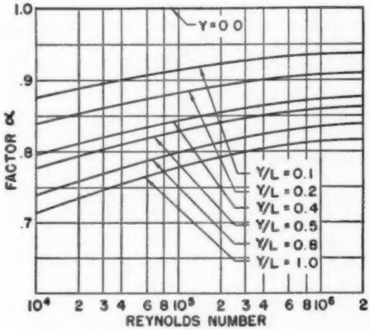


FIG. 13. THE FACTOR α FOR TURBULENT FLOW VS. REYNOLDS NUMBER

section. The curves shown, however, are based upon friction data with length measured to the leading edge of the surface. Therefore heat transfer rates for a surface with length measured from the leading edge were expressed in terms of heat transfer rates measured some distance to the rear of the point where the thermal reactions actually began. The factor α, derived from friction formulae (see Appendix B), has been introduced as the necessary correction factor. Its dependence on the Reynolds number and on the length Y of the heated entrance surface preceding the measuring surface of length L-Y is shown in Fig. 13. Heat transfer surface lengths have been measured from the point where heat transfer (the thermal reaction) actually began and have not included the length of the unheated nose piece. The nose piece, comprised of the support strut and the wood connector strip, was 2 in. long for Tables 1 and 3, 2¾ in. long for Table 2, and 2⁹⁄₁₆ in. long for Table 5 (see Fig. 5).

The data of Tables 1 and 2 show little variation in Y/L. In order to test the correction factor, the mean local heat transfer coefficients listed in Table 3, where Y/L varied from 0.28 to 0.77, were corrected by the appropriate α factor. The results plotted in Fig. 14 show that the agreement of the corrected data with the friction curve lends justification to the use of the α factor.

In Fig. 15, mean local heat transfer coefficients have been plotted as horizontal bars, each of which is so located that its centerline corresponds to the distance from the leading edge of the test plate to the centerline of the test area. The continuous curves (see Appendix B) show the point-to-point variation of the coefficients along the length of the plate. It will be seen that the trend of a coefficient decreasing with length continued up to the fourth test area, where, contrary to theory, the coefficient exceeded that of the third test area. This appeared to be due to the fact that edgewise heat flow from the fourth test area was not fully controlled by the guard area. With the plate reversed in the tunnel and with a turbulent boundary layer flow (velocity in excess for 40 fps) coefficients for the fourth test area, now in the forward position, were substantially identical with coefficients obtained for the first test

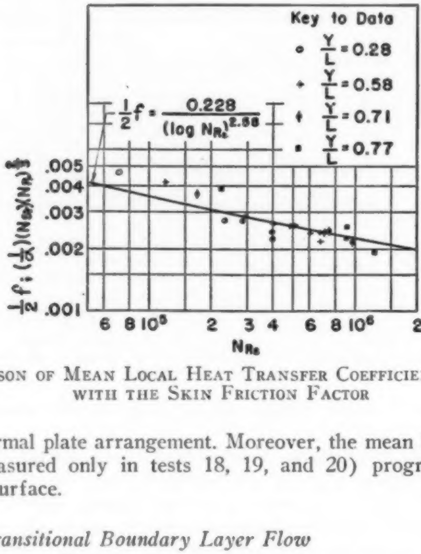


FIG. 14. COMPARISON OF MEAN LOCAL HEAT TRANSFER COEFFICIENTS, MODIFIED BY α, WITH THE SKIN FRICTION FACTOR

area and the normal plate arrangement. Moreover, the mean local heat transfer coefficients (measured only in tests 18, 19, and 20) progressively decreased with length of surface.

Laminar and Transitional Boundary Layer Flow

Velocity and total pressure traverses were made within the boundary layer with the plate in the reversed position. Both the heat transfer data and the total pressure traverses (see Goldstein,[6] Vol. 2, p. 520) indicated that transition took place at approximately 150,000 Reynolds number. (Actually transition takes place over a region rather than at a point.) With this as the critical Reynolds number a transition equation derived from Equations 6 and 8 is:

$$(N_{St})(N_{Pr})^{2/3} = \frac{0.228}{(\log N_{Re})^{2.58}} - \frac{240}{N_{Re}} \qquad (9)$$

This equation is based upon the assumption that, when the boundary layer is partly laminar and partly turbulent, the friction (and consequently the heat transfer) of the turbulent portion of the boundary layer is the same as though the entire boundary layer flow were turbulent.

Corrected for natural convection effects as described later, the data are seen to be in good agreement with the curves of Equations 8 and 9. Appropriate values of α were also used to adjust the test data to the friction curves. It should be pointed out that Table 5 shows that $(N_{St})(N_{Pr})^{2/3}$ based on experimental values of h differs by about 10 percent only from $(N_{St})(N_{Pr})^{2/3}$ based

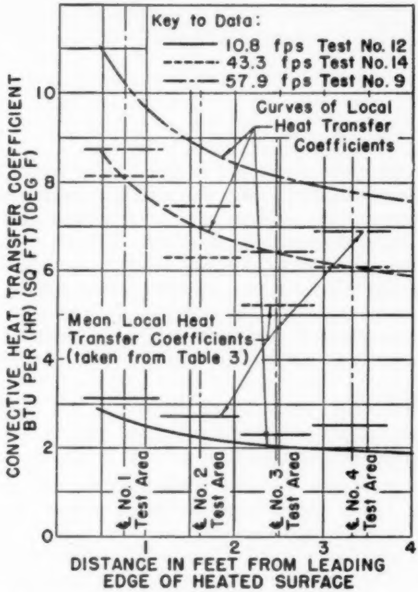


FIG. 15. MEAN LOCAL AND LOCAL HEAT TRANSFER COEFFICIENTS ALONG A FLAT PLATE

on adjusted values. However, the corrections to data for laminar flow were sizable, being of the order of 40 percent downward and 30 percent upward.

Transition of the boundary layer flow along a flat surface from laminar to turbulent velocity distribution is affected by:

1. Initial large scale turbulence or eddying of the air stream.
2. Surface roughness.
3. Leading edge bluntness ratio, i.e., ratio of nose radius to plate thickness (the length of the nose section is also a factor).

Increasing any one of these decreases the Reynolds number at which transition takes place. However, the effect of decreasing any of these factors appears to be limited. The highest Reynolds number at which transition takes place is, according to the literature, approximately 500,000.

No particular significance can be attached to the fact that the transition point in the tests listed in Table 5 was about 150,000 Reynolds number. Since the plate and the various leading edges were of the same order of smoothness and the large scale turbulence constant, the transition point reported here was a function of the bluntness ratio. The ratio was 0.38 for the normal arrangement, 0.19 for the reversed arrangement, and 0.47 for the reversed arrangement with built-up leading edge. With reference to Fig. 12, the intersection of curves A and B at 13,000 Reynolds number suggests that below this value the boundary layer velocity distribution is laminar, irrespective of roughness or other conditions.

Natural Convection Effects

In the course of the tests with laminar flow it was found that natural convection was an important factor. This was indicated by temperature stratification of the air stream at the trailing edge of the plate and by the fact that the test data, unadjusted by appropriate α factors, lay on, rather than below, the theoretical curve. In order to compare the heat transfer data for laminar and transitional boundary layer flow with the friction curves, it was necessary to develop a correction factor for the natural convection effect. Based on the laminar flow data, an approximate relationship was found between the ratio of the natural to forced convection components and the Grashof number,

$$N_{Gr} = \beta\rho^2 g\Delta t\delta'^3/\mu^2 \quad (10)$$

where δ' is the characteristic length dimension and is the computed mean thickness of the laminar boundary layer.

Then to compare these data with the friction curves, adjustments were made both for the effects of the heated entrance length as well as for the effects of natural convection superimposed upon the forced convection. The approximate natural convection component was evaluated and subtracted from the experimental value of h. The remainder was used to evaluate N_{St} and the appropriate α factor was applied. For those tests where both laminar and turbulent flow existed, the corrections for natural convection were applied only to that portion of the coefficient due to the area bounded by the laminar flow.

There appeared to be no natural convection component when the flow was completely turbulent. However, since the lowest velocity with this type of flow was in excess of 10 fps, it is quite possible that such an effect might have appeared at lower velocities. Jurges' [8] data for a flat plate 1.64 ft long with a 1.0 ft long unheated entrance section show this effect. The heat transfer data below 100,000 Reynolds number were 30 to 60 percent above the skin friction curve.

As a check on the symmetry of the plate, one test was made with no air flow. Values of film coefficient of 0.60, 0.58, 0.59, and 0.65 were obtained for test areas 1, 2, 3, and 4 respectively. The average, 0.60, is about 8 percent lower than the value computed from McAdams [9] correlation equation of:

$$h_{nc} = 0.27(\Delta t)^{0.25} \quad (11)$$

The air-to-surface temperature differential based on measurements of the air temperature made in the plane of and 6 in. from the ends of the plate was

34.5 F deg. Two thermocouples were located on the centerline of the plate; two just below the bottom edge.

Effect of Unheated Entrance Length

As stated before, the length of the unheated nose piece was neglected in computing the Reynolds number for the data listed in Tables 1, 2, 3, and 5, because its value was small compared to the length of the heat transfer surfaces. However, several tests were made to study the possible effect of a longer unheated entrance length. In test 11, the first and second test area, and in tests 15 and 16, the first test area, were left unheated. In test 11 the third test area, and in 15 and 16, the second test area were heated to guard the leading edges of the measuring areas. These guard surfaces were treated as heated entrance surfaces and the test data were therefore adjusted by appropriate values of α.

Within the uncertainties connected with the method just described, the entrance length effect was indeterminate. It apparently is small, since definition of L in the Reynolds number as the length of the surface transferring heat placed the data in fair agreement with the friction curve. Jakob and Dow[10] recently investigated this matter by tests with a 1.3 in. diameter cylinder having a heat transfer section 8 in. long. Nose pieces of various lengths up to 12 in. were used. They observed no influence of the unheated entrance length on heat transfer in the laminar boundary layer. Within the limits of their data the effect in the turbulent boundary layer was to increase the average coefficient a maximum of 10 percent. It was stated that their observations probably would apply to the flat plate. This point should not be considered as settled without further study, particularly of the temperature and velocity distributions in the boundary layer.

VII. OTHER PUBLISHED DATA

Non-Uniform Velocity of the Air Stream

Studies of heat transfer from flat surfaces can be classed in two groups. In group one the air stream approaches the test surface with a non-uniform velocity. For example, when the test surface is located in the wall of a duct, a velocity gradient may be developed in the stream before it reaches the test surface (see Fig. 1). In group two the approaching air stream has no previously developed velocity gradient in the immediate vicinity of the test surface. Wind tunnel air streams and free jets belong in this class when the test surface is located in the center of the stream.

The early research on film coefficients undertaken by the A.S.H.V.E. falls in group one. References 11 and 12 were reviewed in the report[13] Heat Transmission Through Windows and Glass Panels. The smooth surface data of reference 11 for 12 x 14 and 6 x 14 in. ducts were re-expressed in terms of the Nusselt and Reynolds numbers. These data were found to correlate when the equivalent pipe diameters were used to evaluate the Reynolds number. These data, corrected for radiation, are plotted in Fig. 16 as curve A, which is a straight line of slope 0.80. Surface conductance coefficients from a third source, reference 14, were computed from overall coefficients for a wall, one

side of which had a glass surface. This surface formed one side of a 12-in. by 8-ft wide duct. These data also lie on curve A. Agreement may be fortuitous, since L/D for reference 14 is low. Smooth surface data from reference 12 also gave a curve of 0.8 slope when the data were corrected for radiation. However, the points lay 25 to 30 percent below curve A. This may be the result of a more fully developed velocity distribution, since the L/D was

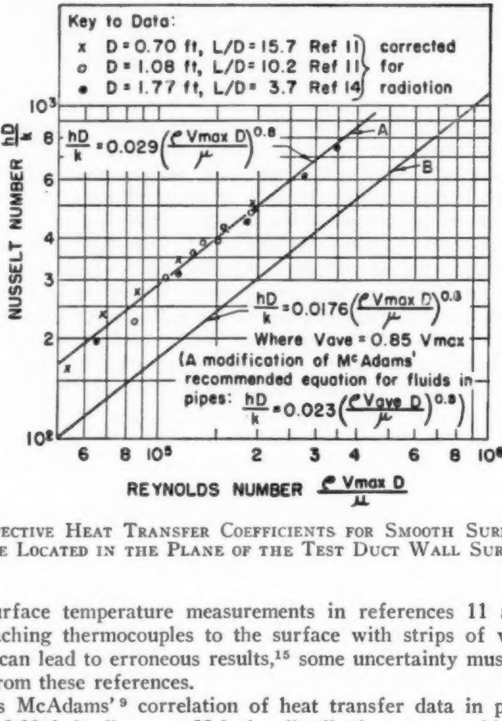


FIG. 16. CONVECTIVE HEAT TRANSFER COEFFICIENTS FOR SMOOTH SURFACES WHICH WERE LOCATED IN THE PLANE OF THE TEST DUCT WALL SURFACE

about 25. Surface temperature measurements in references 11 and 12 were made by attaching thermocouples to the surface with strips of vellum. Since this practice can lead to erroneous results,[15] some uncertainty must be attached to the data from these references.

Curve B is McAdams'[9] correlation of heat transfer data in pipes ranging from 0.04 to 0.33 ft in diameter. Velocity distribution was considered as being fully developed, that is, L/D was in excess of 50 or 60. Curves A and B have identical slopes.

Haucke[16] has studied heat transfer from a pair of steam heated plates 78.7 in. long by 19.7 in. high set 1.89 in. apart. Air was led to the space through a channel also 1.89 in. wide and variable in length up to 78.7 in. The heat transfer coefficient was found to vary inversely as the inlet channel length to the 0.29 power. Though Haucke's results apply to a rather special case, they illustrate the significant effect that a developing velocity field can have on the heat transfer coefficient.

Although the data derived from reference 14 have been compared with other data on the basis of similarity of flow conditions, namely, flow in pipes,

features of the test set-up make these data comparable to flat plate tests in wind tunnels. This comparison is treated in the next section.

In a recent investigation Slegel and Hawkins[17] measured heat transfer coefficients for a smooth heated surface set flush with the wall of a wind tunnel. This surface was 10 in. long; its leading edge was located 31¼ in. from the downstream edge of the honeycomb entrance to the tunnel. Their data, which were expressed by the equation, $N_{Nu} = 0.0492(N_{Pe})^{0.8}$, lie 15 to 20 percent above curve B of Fig. 12. The data lie between 100,000 and 300,000 Reynolds number.

Uniform Velocity of the Air Stream

Several investigators have used wind tunnels to measure heat transfer from flat surfaces. The work of Jurges[8] has been reviewed by Schack,[18] Colburn,[4,5] and Jakob and Dow.[10] Jurges' tests were made with a 1.64-ft square copper plate provided with a 1.0-ft long nose piece. The air stream, 10 in. thick by 23.6 wide, had a uniform velocity to within 1.6 in. of the plate. Jurges' equation for the smooth surface as given by Schack is:

$$h = 0.5V^{0.775} + 0.90e^{-0.183V} \quad \text{. . . . . (12)}$$

where,

e = the base of natural logarithms
V = velocity, feet per second.

The second term of Equation 12 takes into account natural convection. It has a maximum, 0.90, at zero velocity. At 5 fps this term is 20 percent of the first term; at 10 fps it is less than 5 percent. Fig. 17 shows a plot of Jurges' results where h was computed from Equation 12. The fluid properties were evaluated for a temperature of 85 F, the approximate mean temperature prevailing in his tests. As shown by Colburn[4] it is in excellent agreement with Equation 4 at the higher Reynolds numbers.

Elias[19] measured heat transfer from 1.64-ft long copper surface which differed from Jurges' in that the tapered wooden nose piece was 0.33 ft long. The heat transfer coefficient were computed from measurements of the velocity and temperature distributions in the boundary layer, a procedure calling for a high degree of accuracy and good correlation of the two sets of traverses, particularly in the laminar boundary layer. The test points have been carefully scaled from his curve of h/V vs N_{Re} expressed in dimensionless form, and replotted in Fig. 17. A density of 0.072, corresponding approximately to the mean of the surface and air temperatures prevailing in his tests, was used to evaluate N_{St}. The points show considerable scattering at low Reynolds numbers where they indicate a transition from laminar to turbulent boundary layer flows. Elias stated that the transition took place between 140,000 and 185,000. At higher velocities the points are in good agreement with Equation 6. The Reynolds number for the data of both Jurges and Elias was based upon L measured from the beginning of the heat transfer surface.

Fage and Falkner[20] have studied heat transfer in the laminar boundary layer. Platinum foil strips ranging from 0.333 to 1.27 cm in length were electrically heated to maintain a differential of 270 F deg between the air and the strip. Their equation re-expressed in terms of the Stanton number is seen

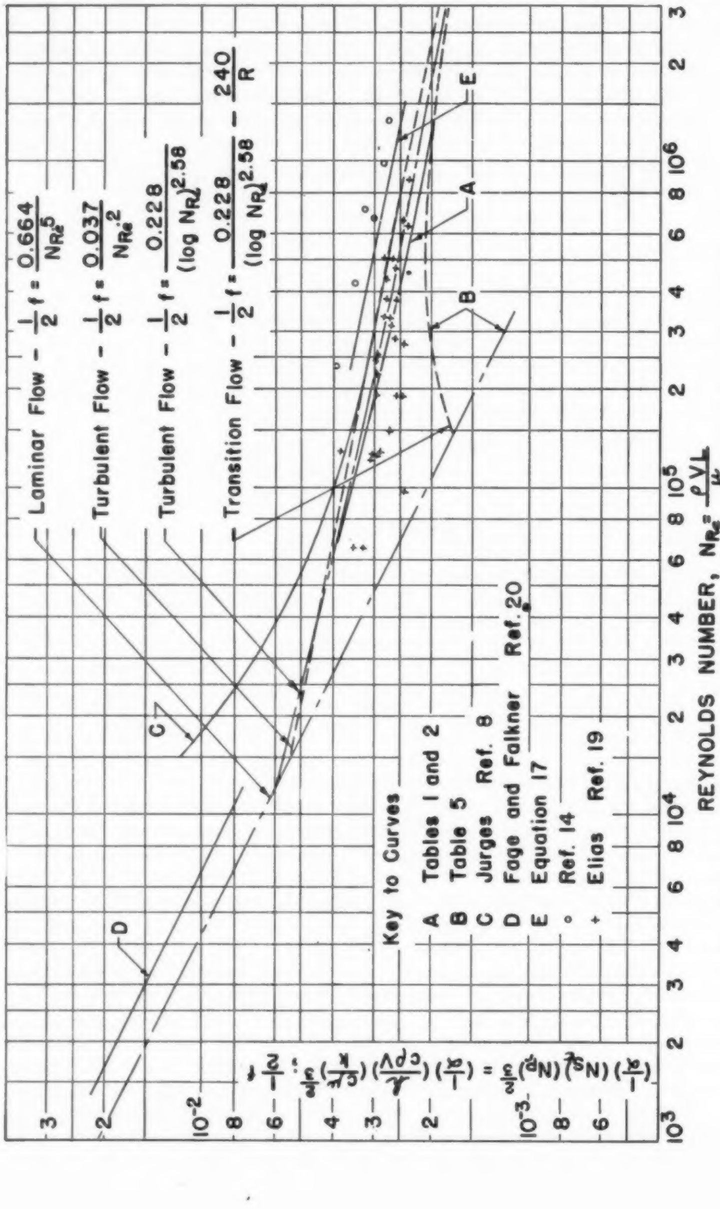


FIG. 17. SKIN FRICTION CURVES AND A COMPARISON OF WIND TUNNEL TESTS OF HEAT TRANSFER FROM SMOOTH FLAT PLATES

in Fig. 17 to lie 25 percent above the friction curve. It is possible that this may be due to natural convection effects.

Jakob and Dow [10] have recently measured heat transfer from a 1.3-in. diameter cylinder placed in the jet from a nozzle with the axis parallel to the flow. They investigated the effects of (1) the shape of the nose piece and (2) the ratio of the length of the unheated nose or *starting length* to the total length. The heat transfer section was 8 in. long. The greatest *starting length* was 12 in. Nose pieces were spherical, ellipsoidal, and conical. Transition took place at 50,000 N_{Re} with the spherical nose and at about 200,000 N_{Re} with the conical nose. Their equations, transposed to the friction form, are, for the laminar region

$$(N_{St})(N_{Pr})^{2/3} = 0.653/(N_{Re})^{0.5} \quad \text{(13)}$$

and for the turbulent region

$$(N_{St})(N_{Pr})^{2/3} = 0.0309/(N_{Re})^{0.2} \quad \text{(14)}$$

Equation 13 is substantially identical with Equation 8; Equation 14 is about 15 percent less than Equation 6. However, Jakob and Dow demonstrate that heat transfer from a cylinder should exceed that from a flat plate of equal area and length. With regard to the analogous skin friction process, Goldstein [6] has stated that a cylinder or similarly shaped body with its axis parallel to the air stream has 10 to 15 percent greater skin friction than a flat surface of equivalent area. He discusses the important effect of large scale eddying or turbulence of the air stream on friction, citing the example of an airship hull model which was tested in a number of wind tunnels in this country and in England. Drag coefficient measurements differed greatly, and, for the same Reynolds number, varied approximately with the magnitude of the wind tunnel turbulence. It is possible that the turbulence of the free jet used on the Jakob-Dow tests was lower than the 1.5 percent turbulence measured in the wind tunnel described earlier in this Bulletin.

A result of the Jakob-Dow study was the development of a relationship between the mean rate of heat transfer for a surface preceded by an unheated entrance section and the mean rate for one of the same total length but heated throughout its length. Equation 14, including this modification factor, becomes:

$$(N_{St})(N_{Pr})^{2/3} = [0.0309/(N_{Re})^{0.2}]\left[1 + 0.4\left(\frac{L_{ST}}{L_{TOT}}\right)^{2.75}\right] \quad \text{(15)}$$

where

L_{TOT} = the total surface length
L_{ST} = the length of the unheated entrance section
N_{Re} is based upon the total surface length, L_{TOT}.

As a maximum the bracketed correction factor reaches 1.4. If this correction be accepted for the flat plate, as suggested by Jakob and Dow, it at first glance appears to be sizable. Actually Equation 6, with N_{Re} based on the length of the heat transfer surface, gives h up to 15 percent greater than Equation 15 modified with the correction factor but with N_{Re} based upon total length. For the test data reported in Tables 1, 2, 3, and 5 correction for the entrance section is insignificant. The data of Table 4 are insufficient to prove the applicability of the Jakob-Dow correction factor to the flat plate.

In their study of finned surfaces, Taylor and Rehbock [21] also measured heat transfer from a plate about 0.5 ft long. They expressed their results, together with Jurges' data, as:

$$h = 0.64\,V^{0.725} \qquad (16)$$

Their own data, which gave a slope of 0.662, were insufficient to correlate and may not have been corrected for radiation. Equation 16 is about 10 percent

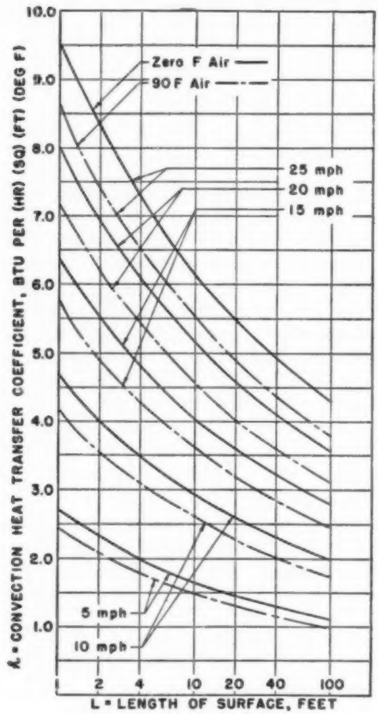


FIG. 18. CONVECTION HEAT TRANSFER COEFFICIENT VS. LENGTH OF SURFACE

above Jurges' equation. Lack of data as to temperatures makes it difficult to make further comparison except to point out that the exponent, 0.662, is unusually low.

As mentioned previously, the test set-up of reference 14 can also be compared with wind tunnel tests. The air stream was directly discharged from a blower plenum into the rectangular channel of which the glass surfaced wall formed one side. The heat transfer length was also the duct length. No straighteners were used in the entrance to this channel. The equation for

curve A of Fig. 16 can be transposed into the form of the friction equation by multiplying the left-hand side by $C/C \times L/D \times \frac{\mu}{\rho V L}$ and the right-hand side by $L/D \times \frac{\mu}{\rho V L}$. This gives

$$\frac{hD}{k} \times \frac{C}{C} \times \frac{L}{D} \times \frac{\mu}{\rho VL} = 0.029 \left(\frac{\rho V}{\mu}\right)^{0.8} \times D^{0.8} \times \frac{L}{D} \times \frac{\mu}{\rho VL}$$

which leads to

$$(N_{St}) \left(\frac{c\mu}{k}\right) = 0.029(1/N_{Re}{}^{0.2}) \left(\frac{L}{D}\right)^{0.2}$$

If 3.7 is now substituted for $\frac{L}{D}$ and $\left(\frac{c\mu}{k}\right)^{2/3}$ for $\left(\frac{c\mu}{k}\right)$ with a corresponding increase in the constant of the right-hand term, the equation ($\frac{c\mu}{k}$ for air equals 0.74) becomes:

$$(N_{St})(N_{Pr})^{2/3} = 0.042/(N_{Re})^{0.2} \quad \text{. . . . . (17)}$$

This is identical in form with Equation 4 and only some 15 percent above Equation 6. This increase could be attributed to turbulence of the entering air stream. This curve is shown as E in Fig. 17 for the range of the data obtained from reference 14.

VIII. APPLICATIONS

Curves for Engineering Use

In using the data presented in this report to calculate film coefficients for conditions common to heating and air conditioning, certain limitations must be recognized. The maximum Reynolds number reached in this test work was approximately 1,250,000, yet in many situations this value is exceeded. For example, the Reynolds number for a 100-ft smooth wall swept by a parallel 15 mph wind is ten times greater. Very high air velocities have been used in studies of heat transfer from airfoils in connection with icing problems, but even in such tests the maximum Reynolds number has not exceeded the value of 1,250,000 because of the small size of the model.

Until the effect of air stream turbulence on heat transfer from smooth flat surfaces has been evaluated, it is suggested that Equation 6 as represented by Curve B of Fig. 12 be used for the computation of coefficients applicable to practical conditions. The use of this curve beyond the range of the experimental work is substantiated by friction data extending to $N_{Re} = 500{,}000{,}000$ and by the heat transfer data reported here over a limited range. Fig. 18 illustrates the effect of surface length and shows the upper and lower limits of the heat transfer coefficient as it is influenced by air temperatures commonly encountered in air conditioning work. Fig. 19 is in a more generalized form that eliminates the need for interpolation. Both curves are based upon a barometric pressure of 14.7 psi and 50 percent relative humidity.

In forced convection heat transfer calculations, it is generally considered more accurate to evaluate the properties of the fluid at the mean of the surface

and fluid temperatures, i.e., the film temperature. This requires a trial and error solution, a procedure which usually is not consistent with the accuracy of other design data. The surface-to-air temperatures encountered in heating work rarely exceed 20 F deg and in such circumstances the simpler method of using the temperature of the free stream introduces negligible error.

In those cases where it becomes necessary to interpolate for length or velocity, or to take into account different air densities, relationships based upon

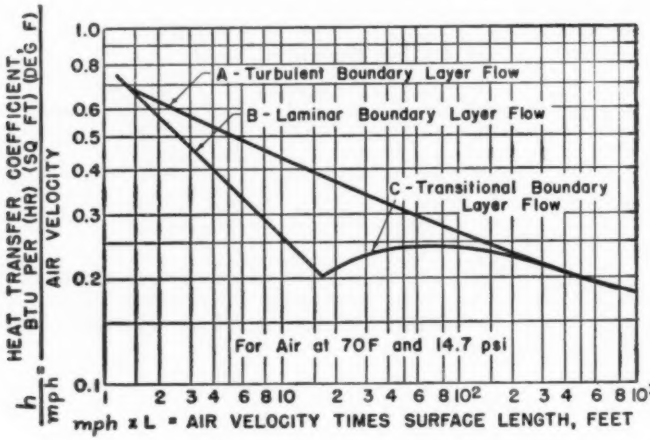


FIG. 19. RELATIONSHIP BETWEEN AVERAGE HEAT TRANSFER COEFFICIENT, AIR VELOCITY, AND LENGTH OF HEATED SURFACE

Equation 4 are recommended for the sake of simplicity. These are:

h varies directly as $V^{0.8}$
h varies directly as $\rho^{0.8}$
h varies inversely as $L^{0.2}$

Heat exchange by radiation must be computed separately. Under certain conditions it can be assumed that the ambient air temperature is also the temperature of the surroundings seen by a surface at which forced convection heat transfer is taking place. Considerable simplification of the calculations can therefore be made by adding the so-called radiation heat transfer coefficient to the convective coefficient. The radiation coefficient is shown for convenience in Fig. 20 as a function of the mean temperature. In certain cases the coefficients are not additive, and in others the radiant heat exchange is regulated by temperatures differing from those governing the convective heat transfer.

At low air velocities the forced convection heat transfer may be augmented by natural convection as suggested earlier in this bulletin. However, the design velocities used in heating and ventilating work are usually high enough so that this effect need not be considered. For low velocities, say under 10 fps,

it is suggested that natural convection be computed by Equation 11 and added to the forced convection coefficient.

It is not yet known to what extent that data given here are applicable to rough surfaces. This roughness may be the general characteristic of a certain type of material or it may be the local roughness caused by window reveals, muntin bars, mortar joints, and the like. This aspect remains to be investigated.

Consideration must also be given to the effect of air flow at an angle to

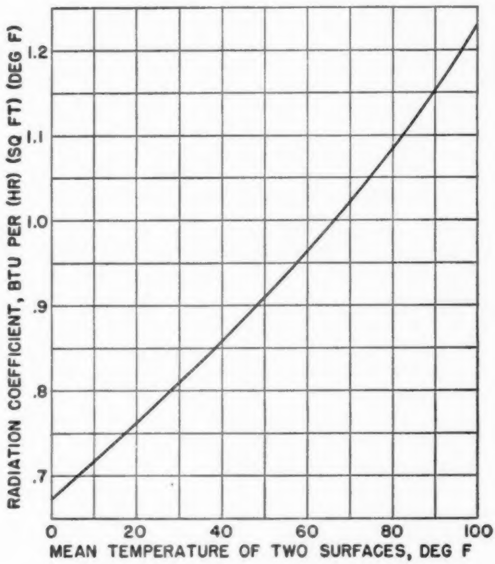


FIG. 20. RADIATION COEFFICIENT (FOR BLACK BODIES) AS A FUNCTION OF MEAN OF TWO SURFACE TEMPERATURES

the surface where higher coefficients are expected and found. For example, the unit rate of heat transfer at the leading edge of an airfoil greatly exceeds that along the chord. This shape, however, approximates a cylinder rather than a large flat surface such as a wall where the effects of angular flow may be limited. The magnitude of large scale turbulence or eddying of the air stream also remains to be evaluated in relation to its effect on the coefficients.

Applications to Glass

The significance of the results of the work herein reported can perhaps be best illustrated by examples (for computation see Appendix C). Table 6 shows the effect of surface length on the computed average overall transmission coefficient, U, of a smooth sheet of glass exposed to the parallel flow of air at 15 mph velocity and zero Fahrenheit degrees. It is assumed that the unit

heat transfer coefficient, h_c, for natural convection on the indoor side of the glass is expressed by the equation, $h_c = 0.27(\Delta t)^{0.25}$, that the indoor temperature is 70 F, and that the glass sees surfaces which are at the ambient air temperature. The width of the air space in double and triple sheets has been taken equal to ½ in. or greater. For closer spacing the overall coefficients are greater than those listed in Table 6.

TABLE 6—CALCULATED OVERALL COEFFICIENTS, *U*, FOR SMOOTH FLAT GLASS
[*Btu per (Hr)(Sq Ft)(F Deg)*]

No. of Sheets	Length of Sheet (Feet)				
	2	4	10	20	100
Single Sheets	1.21	1.18	1.13	1.10	1.04
Double Sheets	0.57	0.56	0.55	0.54	0.53
Triple Sheets	0.37	0.37	0.37	0.36	0.36

Unusual convection currents indoors caused by nearby unit heaters, by radiators, or by air distribution systems will increase these coefficients to a considerable degree. Factors which increase the exchange of heat by radiation, such as panel heating systems, will tend to increase the coefficients. In certain types of buildings or rooms such as greenhouses or solariums, the exchange of heat by radiation between the glass and indoor surroundings is limited. In such cases the effective indoor coefficient is reduced, thereby considerably reducing the values listed in Table 6. The effects of shortwave radiation from the sky and sun can be treated separately by methods outlined in a recent research paper.[22]

As a second example, let it be assumed that a railroad passenger car has 40 ft of glass extending along its side, that its interior temperature is 70 F, and that the outdoor temperature is zero. If it is traveling at 70 mph, the weather side coefficient is found from Equation 6 to be 10.8 Btu per (hour) (square foot) (Fahrenheit degree). The overall coefficient *U* for single and double windows would be 1.33 and 0.60 respectively. It will be recognized that some simplification has been introduced in this example. The film coefficient would be affected by the conditions connected with the actual installation, namely, that the windows do not lie flush with the car surface and that the glass is not a continuous sheet.

IX. SUMMARY

1. The effects of surface length and air velocity on forced convection heat transfer between a smooth flat plate and a parallel stream of air have been measured for Reynolds numbers between 19,000 and 1,200,000.

2. The data for the completely turbulent boundary layer flow were in substantial agreement with a formula based on skin friction measurements of flat plates and were in reasonably close agreement with similar data to be found in the literature.

3. The data for the laminar boundary layer were also placed in agreement with the analogous skin friction formula, after corrections were made for the observed effects of natural convection.

4. Most satisfactory agreement with the friction curves was obtained by defining the length as the length of the heat transfer surface. However, it has been pointed out in the literature that the effect of unheated surface preceding the heat transfer surface can be significant.

5. The air stream turbulence in the wind tunnel was measured and found to be 1.5 percent. Since turbulence is an important factor in heat transfer, it is suggested that future tests include a measurement of the general turbulence by an accepted method.

6. Examples of the application of the results to problems in the field of heating and air conditioning have been given, based on the assumption that the turbulent boundary layer flow is a fair approximation of conditions to be found in practice.

7. Further study is indicated in such matters as air stream turbulence, unheated surface preceding the heat transfer area, and natural convection effects at low velocities.

DEFINITION OF LETTER SYMBOLS

α = a correction factor for heated entrance length.

β = coefficient of expansion; for gases = $1/T$.

δ, δ' = boundary layer thickness, average boundary layer thickness, feet.

μ = absolute fluid viscosity, pounds per (foot)(hour) = centipoises $\times$ 2.42.

ρ = fluid specific weight, pounds per cubic foot.

A = area, square feet.

c = specific heat capacity, Btu per (pound)(Fahrenheit degree).

D = diameter, feet.

f = friction factor, a dimensionless number equal to $\dfrac{2Wg}{A\rho V^2}$.

g = acceleration due to gravity, feet per (hour)(hour).

h = heat transfer coefficient, Btu per (hour)(square foot)(Fahrenheit degree); further described by subscripts c, r, o, i, nc, and fc designating convection, radiation, outdoor, indoor, natural convection, and forced convection, respectively.

k = thermal conductivity, Btu per (hour)(square foot)(Fahrenheit degree per foot).

L = length, feet.

N_{Gr} = Grashof number, a dimensionless number equal to $\dfrac{\beta\rho^2 g\Delta t D^3}{\mu^2}$.

N_{Nu} = Nusselt number, a dimensionless number equal to $\dfrac{hD}{k}$ for pipe flow, $\dfrac{hL}{k}$ for flow along plates.

N_{Pe} = Peclet number, a dimensionless number equal to $N_{Re}N_{Pr}$.

N_{Pr} = Prandtl number, a dimensionless number equal to $\dfrac{c\mu}{k}$.

N_{Re} = Reynolds number, a dimensionless number equal to $\dfrac{\rho VD}{\mu}$ for pipes, $\dfrac{\rho VL}{\mu}$ for flow along flat plates.

N_{St} = Stanton number, a dimensionless number equal to $\frac{h}{c\rho V}$.

t = temperature, Fahrenheit degrees.

Δt = temperature difference, Fahrenheit degrees.

T = temperature, Rankine degrees ($t + 460$).

V = fluid velocity, feet per hour in dimensionless quantities; elsewhere see text or figure.

W = drag force, pounds.

X = length, feet, of unheated surface between leading edge and a particular heat transfer area.

Y = length, feet, of heated surface between leading edge and a particular heat transfer area.

REFERENCES

[1] A.S.H.V.E. RESEARCH PAPER—Forced Convection Heat Transfer Coefficients Along a Flat Surface, by G. V. Parmelee and R. G. Huebscher (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1946, p. 112-116).

[2] Effect of Turbulence in Wind Tunnel Measurements, by H. L. Dryden and A. M. Kuethe (*N.A.C.A.* Report No. 342, U. S. Government Printing Office, Washington, D. C., 1935).

[3] Turbulence Factors of *N.A.C.A.* Wind Tunnels as Determined by Sphere Tests, by R. C. Platt (*N.A.C.A.* Report No. 558, U. S. Government Printing Office, Washington, D. C., 1936).

[4] A Method of Correlating Forced Convection Heat Transfer, and a Comparison with Fluid Friction, by A. P. Colburn (*Transactions American Institute Chemical Engineers*, Vol. 29, 1933, p. 174-210).

[5] Heat Transfer by Natural and Forced Convection, by A. P. Colburn (Bulletin of Purdue University, Engineering Experiment Station, Series No. 84, Lafayette, Ind., January, 1942).

[6] Modern Developments in Fluid Dynamics, S. Goldstein, editor (Vols. 1 and 2, Clarendon Press, Oxford, 1938).

[7] A.S.H.V.E. RESEARCH REPORT No. 1280—A New Friction Chart for Round Ducts, by D. K. Wright, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 303-314).

[8] Heat Transfer to a Plane Wall, by W. Jurges (*Beiheft* 19, Reihe 1, 1924, p. 1).

[9] Heat Transmission, by W. H. McAdams (McGraw-Hill Book Co., Inc., New York, 1942, 2nd ed.).

[10] Heat Transmission From a Cylindrical Surface to Air in Parallel Flow With and Without Unheated Starting Sections, by M. Jakob and W. M. Dow (*A.S.M.E. Transactions*, Vol. 68, No. 2, February, 1946, p. 123-134).

[11] A.S.H.V.E. RESEARCH REPORT No. 895—Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F. C. Houghten and P. McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 301-322).

[12] A.S.H.V.E. RESEARCH REPORT No. 869—Surface Conductances as Affected by Air Velocity, Temperature, and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429-444).

[13] Heat Transmission Through Windows and Glass Panels, by G. V. Parmelee (A.S.H.V.E Laboratory Report to A.S.H.V.E Technical Advisory Committee on Glass, Cleveland, Ohio, December, 1944).

[14] Thermal Transmittance of Glastone, by L. K. Jones (Pittsburgh Testing Laboratory, Pittsburgh, Pa., October 10, 1941).

[15] A.S.H.V.E. RESEARCH REPORT No. 943—Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, by A. P. Kratz and E. L. Broderick (A.S.H.V.E TRANSACTIONS, Vol. 39, 1933, p. 55-62).

[16] The Projection of Heat into Turbulent Air Flow Between Two Flat Parallel Plates, by E. Haucke (*Arch. Warmewirt*, No. 116, 1930, p. 53).

[17] Heat Transfer From a Vertical Plate to an Air Stream, by L. Slegel and G. A.

Hawkins (Bulletin of Purdue University, Engineering Experiment Station, Series No. 97, Lafayette, Ind., May, 1946).

[18] Industrial Heat Transfer, by A. Schack (John Wiley & Sons, Inc., New York, 1933).

[19] Flat Plate Heat Transfer, by F. Elias (Translation in *N.A.C.A.* Technical Memorandum 614, U. S. Government Printing Office, Washington, D. C., April, 1931).

[20] On the Relation Between Heat Transfer and Surface Friction for Laminar Flow, by A. Fage and V. M. Falkner (Aero. Research Committee, R & M No. 1408, 1931-1932).

[21] Rate of Heat Transfer From Finned Metal Surfaces, by C. F. Taylor and A. Rehbock (*N.A.C.A.* Technical Note No. 331, U. S. Government Printing Office, Washington, D. C.)

[22] A.S.H.V.E. RESEARCH REPORT No. 1281—Transmission of Solar Radiation Through Flat Glass Under Summer Conditions, by G. V. Parmelee (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 317-344).

APPENDIX A

REDUCTION OF TEST DATA TO DIMENSIONLESS QUANTITIES

Average Data for Test No. 3 and Computations

1. Heat Input to Plate (Only No. 1 area was tested)

a. Electrical Input (meter reading)	50.88 watts
b. Power taken by meter ($I^2R = 1.20^2 \times 0.226$)	0.33 watts
c. Scale correction (by calibration)	−3.28 watts[a]
d. Corrected input	47.27 watts
e. Equivalent heat input, total	$47.27 \times 3.413 = 161.2$ Btu per hr
f. Area (both sides)	1.454 sq ft
g. Unit rate of heat transfer (item *e* ÷ item *f*)	110.9 Btu per (hr) (sq ft)

2. Temperature

a. Air in front of plate	80.45 F
b. Average plate surface	103.15 F
c. Plate-to-air differential	22.70 F
d. Tunnel wall	80.73 F

3. Heat Exchange by Radiation

a. Mean temperature, plate and tunnel wall	91.94 F
b. Radiation coefficient (Fig. 20)	1.17 Btu per (hr) (sq ft) (F deg)
c. Emissivity factor × angle factor	0.055×1.0
d. Heat lost by radiation $1.17 \times 0.055\,(103.15 - 80.73) =$	1.45 Btu per (hr) (sq ft)

4. Unit Convection Coefficient

$h_c = (110.9 - 1.45) \div (22.70 \times 1) = 4.82$ Btu per (hr) (sq ft) (F deg)

5. Fluid Properties at Mean of Plate and Air Temperatures

Density at 106 grains moisture, 29.24 in. Hg, 91.8 F	= 0.070 lb per cu ft
Specific heat at 106 grains moisture	= 0.243 Btu per (lb) (F deg)
Viscosity	$12.55 \times 10^{-6} \times 3600$ lb per (ft) (hr)

6. Other Data

Velocity at station D	21.6 fps
Length, measured from front edge of heated surface to *rear* edge of test surface (see Fig. 5)	$(4 + 10.25) \div 12 = 1.19$ ft
Y, measured from front edge of heated surface to *front* edge of test surface (see Fig. 5)	$4 \div 12 = 0.33$ ft

7. Dimensionless Numbers

a. $N_{Re} = \dfrac{0.0700 \times 21.6 \times 3600 \times 1.19}{12.55 \times 10^{-6} \times 3600} = 1.43 \times 10^5$

b. N_{Pr} (see McAdams[9]) $= 0.74$

[a] This meter was used only in tests 1 through 5.

c. $N_{St} = \dfrac{4.82}{0.243 \times 0.0700 \times 21.6 \times 3600} = 3.65/10^3$

d. $(N_{St})(N_{Pr})^{2/3} = (3.65/10^3)(0.74)^{2/3} = (3.65/10^3)(0.818) = 2.98/10^3$

e. α (see Fig. 13) (for $N_{Re} = 1.43 \times 10^5$ and $Y/L = 0.28$) $= 0.87$

APPENDIX B

DERIVATION OF THE α FACTOR

The drag *force* of a flat plate moving at constant velocity in a fluid flowing parallel to the length of the plate is found by the following equation:

$$\text{Drag Force} = f A \rho \frac{V^2}{2g} = \text{pounds} \quad \text{(B1)}$$

The drag force due to any particular portion of the plate, for example, a section 2 ft long the forward edge of which is 3 ft from the leading edge of the plate, is computed by finding the drag forces for the 5-ft length and the 3-ft length and subtracting. That is, f is evaluated by means of Equation 3 for N_{Re} based on 5 ft and for N_{Re} based upon 3 ft. The difference, f_5 (5 ft) $- f_3$ (3 ft), is the drag coefficient for the 2-ft section. This, introduced into Equation B1, gives the drag force due to the 2-ft section of the plate located as described.

If then the lengths 5 and 3 are designated respectively as L and Y, as has been done in this report and, the width of the surface as b, giving areas $(L \times b)$, $(Y \times b)$, and $(L - Y)b$, the last being the specific area of interest, then the drag force due to area $(L - Y)b$ is as follows (N_{ReL} and N_{ReY} are evaluated for L and Y respectively):

$$\text{Drag } (L - Y)b = \frac{\rho V^2}{2g}\left[\frac{bL \times 0.455}{(\log N_{ReL})^{2.58}} - \frac{bY \times 0.455}{(\log N_{ReY})^{2.58}}\right] \quad \text{(B2)}$$

The drag coefficient is defined as the drag force divided by $\dfrac{A\rho V^2}{2g}$ and therefore:

$$f_{(L-Y)} = \frac{1}{(L - Y)b} \times \frac{2g}{\rho V^2} \times \frac{\rho V^2}{2g}\left[\frac{bL \times 0.455}{(\log N_{ReL})^{2.58}} - \frac{bY \times 0.455}{(\log N_{ReY})^{2.58}}\right]$$

and

$$f_{(L-Y)} = \frac{0.455}{L - Y}\left[\frac{L}{(\log N_{ReL})^{2.58}} - \frac{Y}{(\log N_{ReY})^{2.58}}\right] \quad \text{(B3a)}$$

modifying Equation B3a as follows:

$$f_{(L-Y)} = \frac{0.455}{L - Y}\left[\frac{L}{(\log N_{ReL})^{2.58}} - \frac{(\log N_{ReL})^{2.58}}{(\log N_{ReL})^{2.58}} \times \frac{Y}{(\log N_{ReY})^{2.58}}\right] \quad \text{(B3b)}$$

whence

$$f_{(L-Y)} = \frac{0.455}{(\log N_{ReL})^{2.58}}\left[\frac{1}{L - Y}\right]\left[L - Y\left(\frac{\log N_{ReL}}{\log N_{ReY}}\right)^{2.58}\right] \quad \text{(B3c)}$$

Since $\quad N_{ReY} = N_{ReL} \times \dfrac{Y}{L}$; then:

$$\left[\frac{\log N_{ReL}}{\log N_{ReY}}\right]^{2.58} = \left[\frac{\log N_{ReL}}{\log\left(N_{ReL} \times \frac{Y}{L}\right)}\right]^{2.58} = \left[\frac{\log N_{ReL}}{\log N_{ReL} + \log \frac{Y}{L}}\right]^{2.58}$$

$$= \left[\frac{1}{1 + \dfrac{\log \frac{Y}{L}}{\log N_{ReL}}}\right]^{2.58}$$

whence, factoring L from the second bracketed term of Equation B3c:

$$f_{(L-Y)} = \frac{0.455}{(\log N_{ReL})^{2.58}}\left[\frac{1}{1 - Y/L}\right]\left[1 - \frac{Y}{L}\left(\frac{1}{1 + \dfrac{\log Y/L}{\log N_{ReL}}}\right)^{2.58}\right] \quad \text{(B4)}$$

Equation B4 can be written:

$$f_{(L-Y)} = \left[\frac{0.455}{(\log N_{ReL})^{2.58}}\right][\alpha] \quad \text{(B5)}$$

where

$$\alpha = \left[\frac{1}{1 - Y/L}\right]\left[1 - \frac{Y}{L}\left(\frac{1}{1 + \frac{\log Y/L}{\log N_{ReL}}}\right)^{2.58}\right] \quad \text{(B6)}$$

Fig. 13 is a plot of α as a function of Y/L and N_{ReL}. If skin friction and heat transfer are related through the term $\frac{0.455}{(\log N_{ReL})^{2.58}}$ then obviously α is equally applicable to both

Equation B6 is evaluated for Y/L equals 1.0 by expressing it as a fraction, differentiating the numerator and denominator, and substituting 1.0 for Y/L. It is found that for this case:

$$\alpha = \frac{\log N_{ReL} - 1.12}{\log N_{ReL}} \quad \text{(B7)}$$

If Equation 4 had been used to derive α instead of Equation 3, for $Y/L = 1.0$, α would be 0.80.

Equation B7 is of value in finding the point-to-point variation of the heat transfer coefficient. If the coefficient at any point L distant from the leading edge is required, Equation 6 is evaluated for L and the required conditions and multiplied by the appropriate value of Equation B7. See Fig. 15 for curves of the point-to-point variation of h.

For the laminar boundary layer flow it can be shown in similar fashion that

$$\alpha = \left[\frac{1}{1 - Y/L}\right][1 - (Y/L)^{\frac{1}{2}}] \quad \text{(B8)}$$

For Y/L equals 1.0, α becomes 0.5.

APPENDIX C

CALCULATION OF OVERALL COEFFICIENTS FOR GLASS

The following assumptions were made with regard to the heat exchange by radiation:

(1) That interior and exterior surfaces seen by the glass were at the ambient air temperature.

(2) That these surfaces were large compared to the glass; therefore, the effective emissivity is that of the glass, 0.93.

(3) That these surfaces totally enclosed the glass; therefore, the effective angle factor is 1.0.

(4) That the effective emissivity for radiant heat exchange between two sheets of glass (infinite parallel planes) becomes 0.87.

With regard to convective heat transfer it is assumed:

(1) That the coefficient for natural convection for flat vertical surfaces is expressed by the equation[9]:

$$h_c = 0.27\ (\Delta t)^{0.25} \quad \text{(C1)}$$

(2) That the coefficient for the conductance across vertical air spaces wider than 1/2 in. is expressed by the equation[22]:

$$h_c = 0.18\ (\Delta t)^{0.25} \quad \text{(C2)}$$

(3) That the glass lies flush with an outer wall and that the flow is parallel with a turbulent boundary layer. Equation 6 expresses the relationship between these flow conditions and the heat transfer coefficient.

The dependence of the film coefficients on temperature requires a trial and error solution to approximate the correct value, particularly with regard to the indoor coefficient.

Under winter conditions as the outdoor temperature falls the glass temperature falls. This increases the natural convection coefficient, which depends upon Δt, and decreases the radiation coefficient, which is a function of mean temperature (see Fig. 20). In summer, both increase as the outdoor temperature increases. For purposes of simplicity in estimating the film coefficients, it will be assumed that there is no temperature gradient across the glass. The error involved in this assumption is negligible in its effect on either the film coefficient or the overall coefficient. The actual solution requires a plotting of curves, which will not be shown here.

Computations for Single Glass

For single glass 4 ft long, 15 mph wind at 0 F, indoor temperature 70 F, glass 1/8 in. thick, conductivity 6.0 Btu per (hr) (sq ft) (F deg per in.):

The outdoor coefficient may be obtained from Equation 6:

$$(N_{St})(N_{Pr})^{2/3} = \frac{0.2275}{(\log N_{Re})^{2.58}}$$

$$\left(\frac{h}{c\rho V}\right)\left(\frac{c\mu}{k}\right)^{2/3} = \frac{0.2275}{\left(\log \frac{\rho VL}{\mu}\right)^{2.58}}$$

whence

$$h_{co} = \frac{0.243 \times 0.0863 \times 5280 \times 15}{(0.74)^{2/3}} \times \frac{0.228}{\left(\log \frac{0.0863 \times 5280 \times 15 \times 4}{10.90 \times 10^{-6} \times 3600}\right)^{2.58}}$$

$$= 4.81$$

Curve plotting (Δt) $(h) = Q$ vs glass temperature shows that the glass temperature is about 16 F. The mean temperature with respect to the outdoors is 8 F, giving a radiation coefficient of 0.70 (Fig. 20). The total outdoor coefficient, h_o, is therefore, $0.93 \times 0.70 + 4.81 = 5.46$ Btu per (hr) (sq ft) (F deg).

The indoor Δt is $(70 - 16)$ or 54 F and the indoor mean temperature is 43 F. The respective convective and radiation coefficients are 0.73 and 0.82 giving a total indoor coefficient, h_i, of 1.55. The overall coefficient U, for the window is:

$$\frac{1}{U} = \frac{1}{h_o} + \frac{l}{k} + \frac{1}{h_i} = \frac{1}{1.55} + \frac{0.125}{6} + \frac{1}{5.46} = 0.645 + 0.021 + 0.183 = 0.849$$

and

$$U = 1 \div 0.849 = 1.18 \text{ Btu per (hr) (sq ft) (F deg).}$$

As a check on heat flow through each film:

16 F drop outdoor air to glass $\times$ 5.46 = 87.5 Btu per (hr) (sq ft)
54 F drop glass to indoor air $\times$ 1.55 = 83.8 Btu per (hr) (sq ft).

This is a satisfactorily close check. It was found that for *the 70 deg differential* h_i varied but little from a value of 1.55. However, for summer conditions of 90 F outdoors and 80 F indoors, $h_i = 1.47$, $h_o = 5.30$, and $U = 1.12$ for the same length of surface and 15 mph.

Computations for Double Glass

The same general procedure was followed with respect to double glass. The differential between the outer glass and the outdoors was found to be about 10 F deg; between the two sheets of glass, about 35 F deg; and between the inner glass and the indoor air, about 25 F deg. The film coefficients respectively, for a 4-ft surface length, are 5.44, 1.13, and 1.48 Btu per (hr) (sq ft) (F deg). The overall coefficient is:

$$\frac{1}{U} = \frac{1}{5.44} + \frac{1}{1.13} + \frac{2 \times 0.125}{6} + \frac{1}{1.48} = 1 \div 1.786$$

whence

$$U = 0.56 \text{ Btu per (hr) (sq ft) (F deg).}$$

The same method has been used to compute the overall coefficients for triple glass and for the example of the railroad car.

1317

No. 1317

INFLUENCE OF GASEOUS RADIATION IN PANEL HEATING†

By F. W. Hutchinson*, Lafayette, Ind.

Introduction

THE exchange of radiant energy between surfaces is usually evaluated on the assumption that the intervening medium is non-absorbing. Actually, both absorption and emission of radiation occur in room air due to the presence of carbon dioxide and of water vapor. The influence of the carbon dioxide is small and that of the water vapor is widely variable, but the combined effect of gaseous radiation in an average room and under average conditions will be shown to be of the order of 10 percent. This value, in itself, is not insignificant, but when metal surfaced reflective materials are considered for use in panel heating, the importance of gaseous radiation is enormously magnified and, as will also be shown, becomes the controlling factor of design for reflective heating systems.

The great practical importance of gaseous radiation in connection with the use of reflective surfaces can be readily visualized by considering the boundary case of an occupant losing heat by radiation to the surfaces of a room 15 ft x 15 ft x 9 ft. If the room surfaces were thermally black, radiant loss from the occupant to his surroundings would occur at the maximum rate of approximately 160 Btu per hr. If the room surfaces were perfect reflectors and if no gaseous radiation were involved, the radiant loss from the occupant would be zero. Thus, it would appear that perfect reflectors would greatly decrease room heating requirements. By equations which are developed later in this paper, it can be shown, however, that the effect of gaseous radiant absorption in such a room would be to dissipate more than 80 percent of the radiant energy emitted by the occupant; thus, even a perfectly reflecting surrounding (which cannot exist in practice) would conserve only 20 percent of body radiant heat

†An analytical paper resulting from a research project on comfort conducted at Purdue University under a grant from the Minneapolis-Honeywell Regulator Co.; this paper was prepared in cooperation with the Housing Research Division of the Purdue Research Foundation: G. Stanley Meikle, Research Director of the Foundation; Carl F. Boester, Housing Executive.
*Professor of Mechanical Engineering, Purdue University, Lafayette, Ind. Member of A.S.H.V.E.
Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Coronado, Calif., June 1947.

loss; in such a case neglect of gaseous absorption would alter the entire concept and lead to a completely ineffective installation.

In an actual room (15 ft x 15 ft x 9 ft) surfaced with material having a reflectivity of 90 percent, radiant dissipation from the occupant would be 95 percent of the maximum possible loss of which 50.3 percent would be by gaseous absorption. In a similar room with reflectivity of 80 percent, the total radiant loss would be in excess of 98 percent of the maximum. Thus in any room of average size, the value to conservation of body heat of surfacing with any commercially available reflective foil would be less than 2 percent; this figure decreases rapidly as room size increases[1].

From the standpoint of total energy requirements of the average non-reflective panel heating system gaseous radiation is relatively unimportant. It is responsible for approximately a 10 percent decrease in the radiant effect of the panel and a consequent increase in the convective fraction of energy dissipation. For a ceiling panel, the convective increase is of the order of 25 percent whereas, for wall or floor panels it is of the order of 14 percent and 9 percent, respectively. The absolute value of the increase of convection is, of course, the same for all panel locations, but the fractional increase is much greater for ceiling panels which normally have a low absolute convection loss.

In industrial installations, or for any local heating panels which are designed to irradiate the subject at a given rate, the average effect of absorption is to necessitate a 10 percent increase in the design area of the panel. It should be noted particularly that this size-increment applies only to local units; general heating panels sized by the usual methods need not be corrected for size, but will have to be corrected for performance differences due to greater air heating effect.

EQUIVALENT GASEOUS RADIATION COEFFICIENT

I. *Water Vapor*. Standard references on heat transfer[2] provide data on the emissive power of water vapor and of carbon dioxide as functions of the vapor pressure, the gas temperature and the mean length of path through the gas. For water vapor at temperatures such as occur in panel heating, the emissive power can be expressed by the following equation:

$$E = aTg^{3.64}$$

where

E = emissive power of water vapor, Btu per (hr) (sq ft)
Tg = absolute temperature of gas, deg
a = a coefficient which varies as a function of the product PL
P = partial pressure of the water vapor expressed as a fraction of atmosphere
L = beam length, varying from ⅔ of the diameter for a sphere to 1.3 times the ceiling height for a room of proportions $1 \times 2 \times 6$ in which radiation is to one of the large faces.[3]

For rooms of average size and shape, with radiation considered as occurring from a ceiling or a floor panel, the value of L can conveniently, and with ac-

[1]Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, pp. 35-44.)
[2]Heat Transmission, by McAdams.
[3]Heat Transmission, by McAdams, p. 63.

curacy, be taken as equal to the ceiling height. The partial pressure of water vapor in room air is fixed when the absolute humidity is known; hence can be expressed as a function of room air temperature and relative humidity as two conveniently measurable variables.

In evaluating radiant exchange between surfaces separated by a non-absorbing medium, it has been found convenient to use an equivalent film coefficient for radiation, h_r, defined as,

$$h_r = \frac{0.172\left[\left(\frac{T_h}{100}\right)^4 - \left(\frac{T_c}{100}\right)^4\right]}{t_h - t_c} \quad \ldots \quad (1)$$

where

h_r = equivalent film coefficient for radiation, Btu per (hr) (sq ft) (F deg)
T_h = absolute (F) temperature of *hot* surface
T_c = absolute (F) temperature of *cold* surface
t_h = temperature (F) of *hot* surface
t_c = temperature (F) of *cold* surface

A similar equivalent coefficient for radiation to or from water vapor is,

$$h_w = \frac{a\,(T_h^{3.54} - T_c^{3.54})}{t_h - t_c} \quad \ldots \quad (2)$$

where

h_w = equivalent film coefficient for radiation to or from water vapor, Btu per (hr) (sq ft) (F deg)

Fig. 1 is a graphical solution of Equation 2. Use of the graph can best be explained by means of an example: In a room with air at 72 F, 50 percent relative humidity, to evaluate the absorption by the water vapor of radiation from a surface at 83 F; ceiling height is 9 ft.

Solution: Enter the upper right quadrant at 72 F and move horizontally left to intersect the 50 percent RH line (see dashed example on graph) then rise to the curve for 9 ft ceiling height, move horizontally left to the directrix and drop vertically to the lower left quadrant. Now re-enter the chart in the lower right quadrant at the 83 F surface temperature, rise to 72 F air temperature and move horizontally left to intersect the previously established vertical at a value of h_w=0.086. Thus the transfer rate to the gas by radiation is 0.086 (83 — 72) = 0.95 Btu per sq ft.

II. *Carbon Dioxide*. The emissive power for carbon dioxide is given by an equation similar to that for water vapor, but with an exponent of 5.3 and a different functional relationship between the coefficient "*a*" and the value of *PL* for CO_2. The partial pressure of CO_2 in outside air is 0.0003 and in the air of an occupied space its partial pressure reaches an equilibrium value dependent on the ventilation rate. Taking 0.6 cfh as the CO_2 output of an occupant and taking 10 cfm per occupant as the minimum outside air used for ventilation, the equilibrium CO_2 concentration, *X*, is given by

$$0.0003 \times 10 \times 60 + 0.6 = 60 \times 10X$$
$$X = 0.0013 \text{ (or 0.13 percent)}$$

Taking $L = 10$, "*a*" is 7.6×10^{-15} and the value of the equivalent coefficient for exchange between a 120 F panel and 60 F gas is approximately 0.025 Btu per

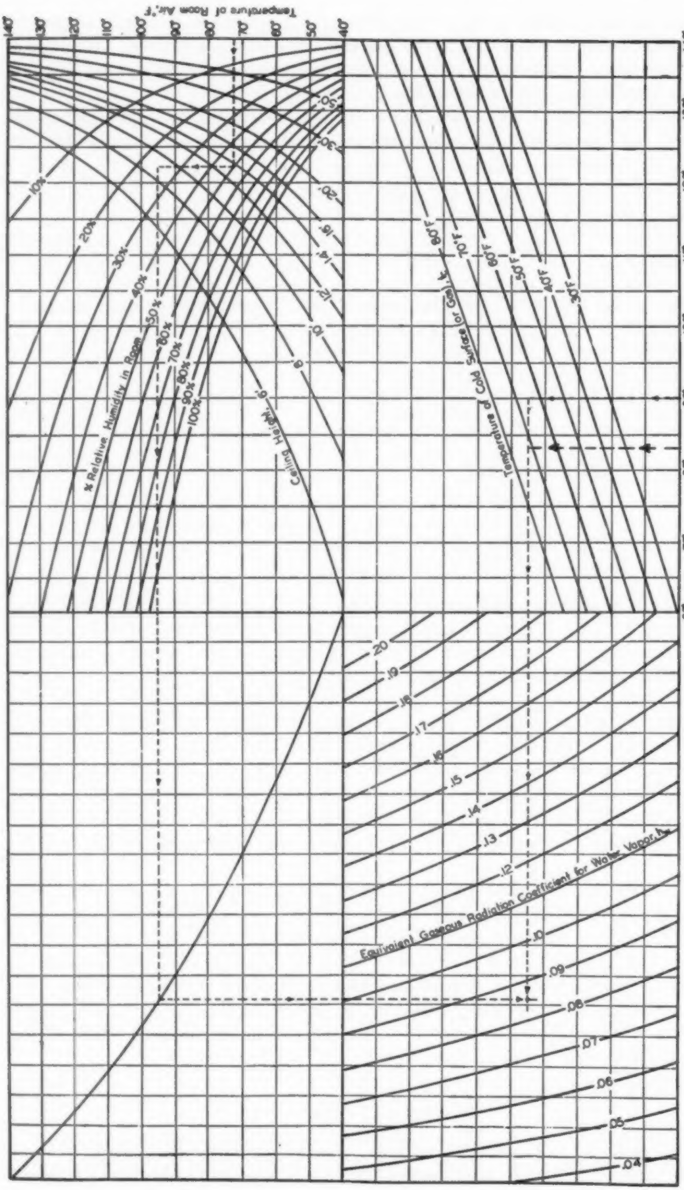


FIG. 1. GRAPHICAL SOLUTION OF EQUATION 2 FOR DETERMINING EQUIVALENT COEFFICIENT FOR RADIATION TO OR FROM WATER VAPOR

(hr) (sq ft) (F deg). Values of this coefficient greater than 0.025 are unlikely to occur in panel heating practice and since the coefficient is small with respect to usual values of the water vapor coefficient, it is suggested that a fixed conservative value of 0.02 be used for all average cases.

III. *Total Gaseous Radiation.* An equivalent gaseous radiation coefficient, h_g, can now be defined by the equation,

$$h_g = h_w + 0.02 \quad \quad (3)$$

where

h_g = equivalent gaseous radiation coefficient, Btu per (hr) (sq ft) (F deg)
h_w = equivalent water vapor radiation coefficient, Btu per (hr) (sq ft) (F deg)

where h_w is determined by Fig. 1. For many practical problems, the coefficient can be further simplified by selecting representative average temperatures and thereafter neglecting the effect on h_g of the temperature. Similarly, *standard* room conditions can be selected, the ceiling height fixed and a single value of h_g thereby obtained which will be applicable to most problems. For room air at 72 F, 50 percent relative humidity with 9 ft ceiling and the temperatures arbitrarily fixed at 65 F and 90 F the value of h_w from the graph (see dashed example line in Fig. 1) is 0.086 and the corresponding value of the total gas coefficient h_g is,

$$h_g = 0.11$$

When the statement of a problem does not permit exact evaluation of the conditions of gaseous radiant exchange, it is recommended that the value of the coefficient given previously be used.

In similar manner, a fixed value of the equivalent radiation coefficient for transfer through a non-absorbing medium is frequently used. For the temperature range common in panel heating, the value of this constant has been taken as 1.08. Thus for an *average* room under *average* panel heating conditions, the effect of gaseous absorption is to reduce the energy exchange by radiation between the heating panel and the surroundings by

$$0.11/1.08 = 10 \text{ percent}$$

For convenience in the analyses of the following section, the difference between unity and the fraction of absorbed radiant energy will be defined as the transmissivity, ν; for standard conditions 90 percent of the energy emitted is transmitted so the corresponding value of is 0.90.

BASIC EXCHANGE EQUATIONS WITH GASEOUS RADIATION CONSIDERED

If all surfaces of practical importance were thermally *black bodies,* the gaseous radiation coefficient from the preceding section could be used as a direct indication of the importance of transfer between a surface and the ambient air. In many cases, however, thermally *gray* surfaces, and metallic surfaces of high reflectivity, must be considered. When these conditions occur, the importance of gaseous absorption increases at a very rapid rate since the average unit of thermal energy emitted by the source will traverse the space between source and receiver more than once. Hence equations must be established from which the additional increments of gaseous absorption, associated with interreflections and re-reflections in the system composed of source and receiving surfaces, can be evaluated.

As in most practical problems, a satisfactory rational solution can only be obtained in terms of a simplifying idealization. The variety of shapes and surface characteristics of real rooms are so great, and the possible combinations so complex, that no accurate generalized solution can be found. Recourse is had, therefore, to an idealization which will give an approximation to results for the general case and an exact solution of certain particularly important boundary cases. Consider, for example, energy exchange between the shells of two concentric spheres: As the inner sphere becomes small with respect to the outer, the ratio of surface areas approaches zero and the system becomes equivalent to that represented by an element of area, dA_1, of an enclosure; the quantity of energy initially emitted by dA_1 and later re-absorbed is negligible, but the influence of reflectivity of the surroundings on total gaseous absorption remains great. If, however, the inner sphere becomes large with respect to the outer the fraction of energy reflected from A_2 and returning to A_1 approaches unity; this system becomes similar to one involving infinite parallel planes.

For values of the area ratio between zero and unity, the concentric spheres can be shown to form a system similar to that of concentric cylinders of infinite length and they also form a system from which one can approximate the exchange of body heat between an occupant and a uniform surrounding. Thus an exact rational solution for the idealization of concentric spheres will be of value in approximating a solution to many practical problems. As a first step toward development of such a solution, it may be noted[4] that (A_1/A_2) of the energy reflected from A_2 is returned in the direction of the inner sphere.

I. To determine the net fraction, F_1, of *black body* radiation initially emitted by the outside surface of a sphere of area A_1 and either absorbed by the inside surface of a larger concentric sphere of surface area A_2 or absorbed by the water vapor and carbon dioxide contained in the air in the space between the two spheres. Thus,

Net initially emitted radiant energy leaving A_1 is the total emission from A_1 minus energy re-absorbed after reflection or,

$$q_1 = q_e - q_r$$

now, q_e, the emissive power $= a_1 \sigma T_1^4 = E_1$

where

a_1 = emissivity of surface A_1
a_2 = emissivity of surface A_2
$\sigma = 0.173 \times 10^{-8}$

Energy striking A_2 for the first time and returned in the direction of A_1 is,

$$E_1 A_1 \nu^2 (1 - a_2) (A_1/A_2)$$

where

ν = transmissivity of the room air to radiant energy's difference between unity and the gaseous adsorption fraction.

---

[4]Heat Transfer Notes, by Boelter, Cherry and Johnson. (University of California Press, Chapter XVIII, section 11d.)

Energy striking A_2 for the first time and reflected in the direction of A_1 is,

$$E_1 A_1 \nu^2 (1 - a_2) (1 - A_1/A_2)$$

Energy striking A_2 for the second time after one reflection from A_2 and then returned in the direction of A_1 is,

$$E_1 A_1 \nu^3 (1 - a_2)^2 (1 - A_1/A_2) (A_1/A_2)$$

Energy striking A_2 for the second time after one reflection from A_2 and then reflected in the direction of A_2 is,

$$E_1 A_1 \nu^3 (1 - a_2)^2 (1 - A_1/A_2)^2$$

The total energy returned in direction of A_1 for the first time is then,

$$E_1 A_1 \nu^2 (1 - a_2) (A_1/A_2) \Big[1 + \nu (1 - a_2) (1 - A_1/A_2) + \ldots + \nu^n (1 - a_2)^n (1 - A_1/A_2)^n \Big] = \frac{E_1 A_1 \nu^2 (1 - a_2) (A_1/A_2)}{1 - \nu (1 - a_2) (1 - A_1/A_2)}$$

The energy initially emitted by A_1 and leaving for a second time is,

$$\frac{E_1 A_1 \nu^2 (1 - a_2) (A_1/A_2) (1 - a_1)}{1 - \nu (1 - a_2) (1 - A_1/A_2)}$$

and the energy returned in the direction of A_1 for the second time is,

$$\frac{E_1 A_1 \nu^2 (1 - a_2) (A_1/A_2)}{1 - \nu (1 - a_2) (1 - A_1/A_2)} \bullet \left[\frac{\nu^2 (1 - a_1) (1 - a_2) (A_1/A_2)}{1 - \nu (1 - a_2) (1 - A_1/A_2)}\right]$$

so the energy returned in the direction of A_1 for the nth time is,

$$\frac{E_1 A_1 \nu^2 (1 - a_2) (A_1/A_2)}{1 - \nu (1 - a_2) (1 - A_1/A_2)} \bullet \left[\frac{\nu^2 (1 - a_1) (1 - a_2) (A_1/A_2)}{1 - \nu (1 - a_2) (1 - A_1/A_2)}\right]^{n-1}$$

Thus the *total* energy initially emitted by A_1 and returned in the direction of A_1 after multiple reflections is,

$$\frac{E_1 A_1 \nu^2 (1 - a_2) (A_1/A_2)}{1 - \nu (1 - a_2) (1 - A_1/A_2)} \bullet \left\{\frac{1}{1 - \dfrac{\nu^2 (1 - a_1) (1 - a_2) (A_1/A_2)}{1 - \nu (1 - a_2) (1 - A_1/A_2)}}\right\}$$

or

$$\frac{E_1 A_1 \nu^2 (1 - a_2) (A_1/A_2)}{1 - \nu (1 - a_2) (1 - A_1/A_2) - \nu^2 (1 - a_1) (1 - a_2) (A_1/A_2)}$$

and the energy re-absorbed is a_1 times the above quantity. Then the net fraction of black body energy leaving A_1 is,

$$F_1 = a_1 - \frac{a_1^2 \nu^2 (1 - a_2) (A_1/A_2)}{1 - \nu (1 - a_2) (1 - A_1/A_2) - \nu^2 (1 - a_1) (1 - a_2) (A_1/A_2)}$$

$$F_1 = \frac{a_1 \Big\{(1 - \nu + a_2 \nu) \Big[1 + \nu (A_1/A_2)\Big] - a_2 \nu (A_1/A_2)\Big\}}{1 - \nu (1 - a_2) \Big[1 - (A_1/A_2) + \nu (A_1/A_2) - a_1 \nu (A_1/A_2)\Big]} \quad \ldots\ldots \quad (4)$$

II. To determine the net fraction, F_r, of *black body* radiation initially emitted by the outside surface of a sphere of area A_1 and absorbed by the inside surface of a larger concentric sphere of surface area A_2.

Total energy received at A_2 of that initially emitted by A_1

= (Direct from A_1 with zero to infinite inter-reflections at A_2)
+ (Second time from A_2 with zero to infinite inter-reflections at A_2)
+ ——— + ——— +
+ (Infinite times from A_2 with zero to infinite inter-reflections at A_2)
= (A) + (B) + ——— + (N)

By inspection (following the procedure of the derivation in part I),

$$(A) = E_1 A_1 \nu a_2 + \frac{E_1 A_1 \nu^2 (1 - a_2) \left[1 - (A_1/A_2)\right] a_2}{1 - (1 - a_2) \nu \left[1 - (A_1/A_2)\right]}$$

$$(B) = (A) \cdot \left\{ \frac{\nu^2 (1 - a_2)(1 - a_1)(A_1/A_2)}{1 - (1 - a_2) \nu \left[1 - (A_1/A_2)\right]} \right\}^2$$

$$(N) = (A) \left\{ \text{———} \right\}^{n-1}$$

The total energy received at A_2 is then,

$$\frac{E_1 A_1 \nu a_2 + \dfrac{E_1 A_1 \nu^2 (1 - a_2) \left[1 - (A_1/A_2)\right] a_2}{1 - (1 - a_2) \nu \left[1 - (A_1/A_2)\right]}}{1 - \dfrac{\nu^2 (1 - a_1)(1 - a_2)(A_1/A_2)}{1 - (1 - a_2) \nu \left[1 - (A_1/A_2)\right]}}$$

and the net fraction of black body radiation emitted by A_1 which is absorbed by A_2 is,

$$F_r = \frac{a_1 a_2 \nu}{1 - \nu (1 - a_2) \left[1 - (A_1/A_2) + \nu (A_1/A_2) - a_1 \nu (A_1/A_2)\right]} \quad . \; . \; . \; . \; (5)$$

III. To determine the net fraction, F_a, of *black body* radiation initially emitted by the outside surface of sphere A_1 and absorbed by the water vapor and carbon dioxide in the air in the space external to this sphere but inside a larger concentric sphere.

The fraction of black body energy leaving A_1, (I), less the fraction received at A_2, (II), must be equal to the fraction absorbed, thus,

$$Fa = F_1 - F_r = \frac{a_1 \left\{ (1 - \nu + a_2\nu) \left[1 + \nu (A_1/A_2)\right] - a_2\nu (A_1/A_2) - a_2\nu \right\}}{1 - \nu (1 - a_2) \left[1 - (A_1/A_2) + \nu (A_1/A_2) - a_1\nu (A_1/A_2)\right]} \quad . \; (6)$$

IV. Check on the three basic equations:—

1. If gaseous absorption is neglected $\nu = 1$ and the equations become:

$$F_1 = F_r = \frac{a_1 a_2}{\frac{1}{a_1} + (A_1/A_2)\left[\frac{1}{a_2} - 1\right]} \qquad (7)$$

which is the shape factor for concentric spheres.[5]

2. If the inner sphere is very small with respect to the outer, the ratio A_1/A_2 approaches zero and

$$F_1 = \frac{a_1 (1 - \nu + a_2 \nu)}{1 - \nu (1 - a_2)} \qquad (8a)$$

$$F_r = \frac{a_1 a_2 \nu}{1 - \nu (1 - a_2)} \qquad (8b)$$

$$Fa = \frac{a_1 (1 - \nu)}{1 - \nu (1 - a_2)} \qquad (8c)$$

3. If the inner sphere is large with respect to the outer, the ratio A_1/A_2 approaches unity and

$$F_1 = \frac{a_1 (1 - \nu^2 + a_2 \nu^2)}{1 - \nu^2 (1 - a_2)(1 - a_1)} \qquad (9a)$$

$$F_r = \frac{a_1 a_2 \nu}{1 - \nu^2 (1 - a_2)(1 - a_1)} \qquad (9b)$$

$$Fa = \frac{a_1 (1 - \nu^2 - a_2 \nu^2 - a_2 \nu)}{1 - \nu^2 (1 - a_1)(1 - a_2)} \qquad (9c)$$

APPLICATIONS OF THE EXCHANGE EQUATIONS

I. Influence of gaseous absorption in a room in which all surfaces are assumed perfect reflectors ($a_2 = 0$) and the occupant is considered to have an emissivity of 0.9:—To calculate the percentage of radiant energy emitted by the occupant and absorbed by the gas,

1. If there were no gaseous absorption ($\nu = 1.0$) the energy loss from the occupant would be (Equation 7),

$$F_1 = 0$$

2. With gaseous absorption, the loss is given by Equation 4 and F is equal to F_1. The magnitude of the loss will depend on room size since for larger rooms the average unit of radiation will have to undergo a greater number of inter-reflections between room surfaces before being returned to the occupant. For a room 15 ft x 15 ft x 9 ft (having 990 sq ft of surface area): By Equation 4, $F_1 = 0.9 \times 0.802$, or 80 percent is absorbed.

II. Radiant loss from occupant to an enclosure having a reflectivity of 90 percent. (Size 15 ft x 15 ft x 9 ft).

By Equation 6, $F_a = 0.9 \times 0.503$
By Equation 5, $F_r = 0.9 \times 0.445$
Loss $= 0.9 \times 0.948$ or 95 percent of emitted energy

III. Radiant loss from occupant to an enclosure (15 ft x 15 ft x 9 ft) having a reflectivity of 80 percent.

By Equation 4, $F_1 = 0.9 \times 0.98$ or 98 percent of emitted energy.

[5]Heat Transmission, by McAdams (1st Edition, p. 54, case 5).

This case represents conditions comparable to the best obtainable with reflective surfacings. Although reflectivities in excess of 80 percent are available, such materials are unsuitable for use on floors and even if all walls and the ceiling (ignoring, for the moment, windows) were covered with such surfacing, the *equivalent* reflectivity of the surrounding surfaces would, in almost every case, be less than 80 percent.

SUMMARY AND CONCLUSION

It is a popular misconception that separating distance (aside from its effect on shape factor) does not affect the rate of radiant exchange between two surfaces. Actually, radiant exchange decreases as distance increases because of absorption by the carbon dioxide and water vapor present in the intervening air space. A graphical solution is presented from which the equivalent coefficient for gaseous radiant exchange can be evaluated as a function of separating distance, vapor pressure (expressed in terms of room air temperature and of relative humidity) and surface temperatures. Equations are developed to permit evaluation of radiant exchange between certain systems of surfaces when they are reflective and are separated by an absorbing medium.

Gaseous radiation does not appreciably affect either the panel size or panel rating for an ordinary panel heating system, but it does reduce the effectiveness of local (direct transfer) panels by about 10 percent. Because of cumulative absorption as associated with multiple reflections, gaseous radiation is responsible for reducing the effectiveness of reflective surfaces (when used as room surfacing in an attempt to reduce radiant body heat loss) to a negligible value; thus use of commercial foil for surfacing a room larger than 15 ft x 15 ft x 9 ft would conserve only about 2 percent (3 Btu per hr) of body heat.

1318

No. 1318

CONDITIONS FOR COMFORT

By CHARLES S. LEOPOLD*, PHILADELPHIA, PA.

THERE are many definitions of comfort, but the one chosen for this discussion is: The absence of discomfort or annoyances due to temperature and atmospheric effects indoors. Considerations of health, deliberate stimulation, and lowering of activity are not primary factors in optimum comfort or minimum discomfort as thus defined.

For rooms of multiple occupancy, and particularly for desk work, it follows that the optimum conditions are those in which the occupants are not conscious of discomforts or annoyances due to temperature and atmosphere. This definition would include the effects of radiation, air temperature, water vapor, odor, and air motion. Active conditions such as *comfortably warm* and *comfortably cool* are not acceptable to a group.

For actual installations, the criterion should be: What conditions are acceptable to the greatest number who do not have to pay for the maintenance of these conditions, and who are unaware that a test is being conducted? For house heating, if the master of the house must pay a high fuel bill, he quite likely will find an ***optimum*** which compromises with his pocketbook. His guests will be just plain cold. In any building, if the subjects know that a test is being conducted, they tend to interpret their sensations and are likely to report how they think they should feel. Many people seem to believe it is a virtue to endure cold, indoors.

There are two buildings of massive construction in Washington in which air conditioning was installed about ten years ago. In each, air is supplied by large systems, and the final control for each room is obtained by reheat controlled by a thermostat with a temptingly large adjusting knob. The moisture of the conditioned air is controlled to give approximately 50 percent relative humidity in summer and 30 percent relative humidity in winter. The total occupancy (ages 18 to 80) is over 2000. Air motion is moderate. One building has radiators beneath the windows; the other depends on the air system for all heating requirements.

---

*Consulting Engineer. Member of A.S.H.V.E.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Coronado, Calif., June 1947.

A census of the control setting at any time, summer or winter, will show upwards of 90 percent of the room thermostats set for 75 F to 76 F and maintaining 74 F to 77 F.

A large office building, near Washington, has a population of 25,000 to 35,000.

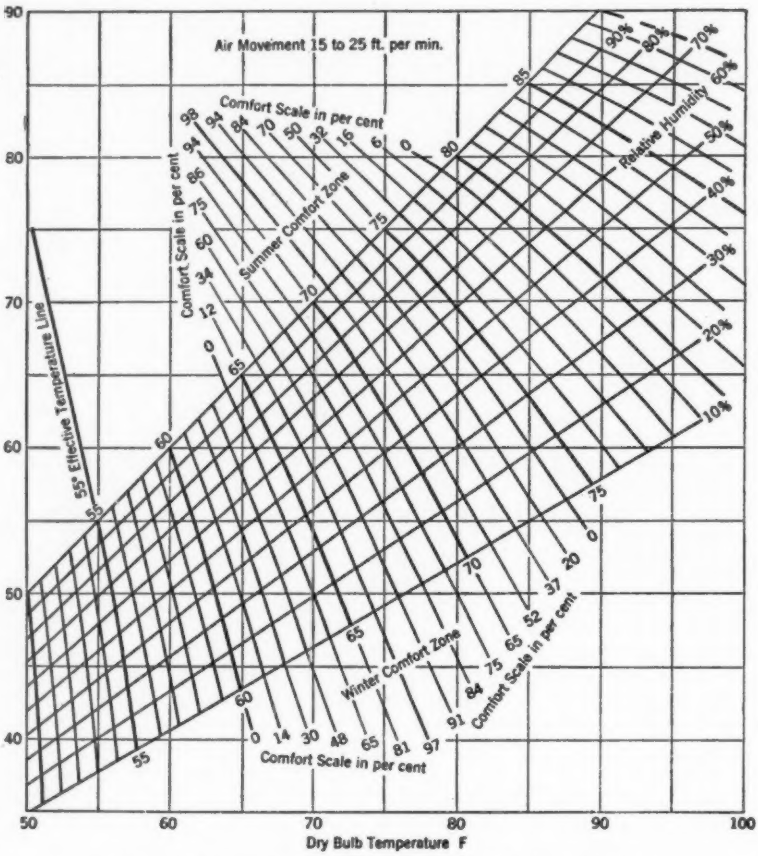


FIG. 1. ORIGINAL A.S.H.V.E. COMFORT CHART (1929).

A thermostat setting of about 75 F appears to be the best for both interior and exterior zones year-round.

An insurance building in the Middle West, with approximately 1000 employees, operates at about 76 F, summer and winter. The radiation, due to transmission through walls and windows, is balanced at the wall. In this case, the

higher temperature is probably required because the cooling air is uniformly introduced through a perforated ceiling, which results in a year-round lower mean radiant temperature of the room.

Two large office buildings in Texas operate at approximately 77 F throughout the year. These buildings have complete zone control and are extensively subdivided. The desire for slightly higher temperature is probably due, in part, to acclimatization and, in part, to working slightly on the warm side to compensate for the absence of individual controls in small offices.

In all but the first example, conditions were finally set by the operating engineer faced with the problem of satisfying the most people. In the first example, they are set by the occupants themselves.

Mass determinations, such as the foregoing, lack the accuracy of well conducted laboratory experiments but they have the great virtue of numbers and the absence of conscious psychological factors or systematic errors due to the emloyment of a few *trained* subjects.

The thought that an appreciable percentage of people will be dissatisfied, regardless of conditions, is not supported by observation of systems properly designed and operated. In drawing conclusions from field experience, however, it is advisable to exclude from the data systems in which:

1. The draft problem has not been thoroughly solved.
2. There are pronounced unbalanced radiant effects.
3. Constant conditions cannot be maintained throughout the day because of limited capacity.

If laboratory work does not agree with the mass experience, we should then question either:

1. The laboratory technique, or
2. The mental process whereby the results of the laboratory are applied to everyday living.

The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, for many years has contributed valuable laboratory data on this subject. Unfortunately, the A.S.H.V.E. Comfort Chart, rather than the valuable and voluminous A.S.H.V.E. publications, is the major contact with many engineers.

Fig. 1 shows the chart as originally published in 1929. It contains A.S.H.V.E. laboratory data as found, plus a concept of a *comfort zone*. The indicated zone included all conditions from a maximum to zero percent comfortable and, therefore, had a definite meaning as it indicated the total range of conditions in which some people were comfortable. This chart confirms the summer optimum of approximately 76 F and 50 percent relative humidity but does not agree as to winter optimum. The optimum condition is somewhat subject to seasonal acclimatization which tends to explain geographical deviation. In general, the latter appears to be a rather small effect.

Fig. 2, published in 1932, stresses and redefines the *comfort zone* as limited to the area bounded by 30 percent and 70 percent relative humidity, and to 50 percent of the subjects comfortable on the cool side and 50 percent of the subjects comfortable on the warm side. This zone is purely arbitrary. Certainly, a condition in which 50 percent of the people are uncomfortable is a poor type of comfort. In producing this chart, the scale showing the percent of people

uncomfortable was omitted, thus tending to obscure the fallacy of the concept. The *comfort zone*, as here defined, did great service as propaganda but was a poor engineering approach, since it tended to obscure the real problem.

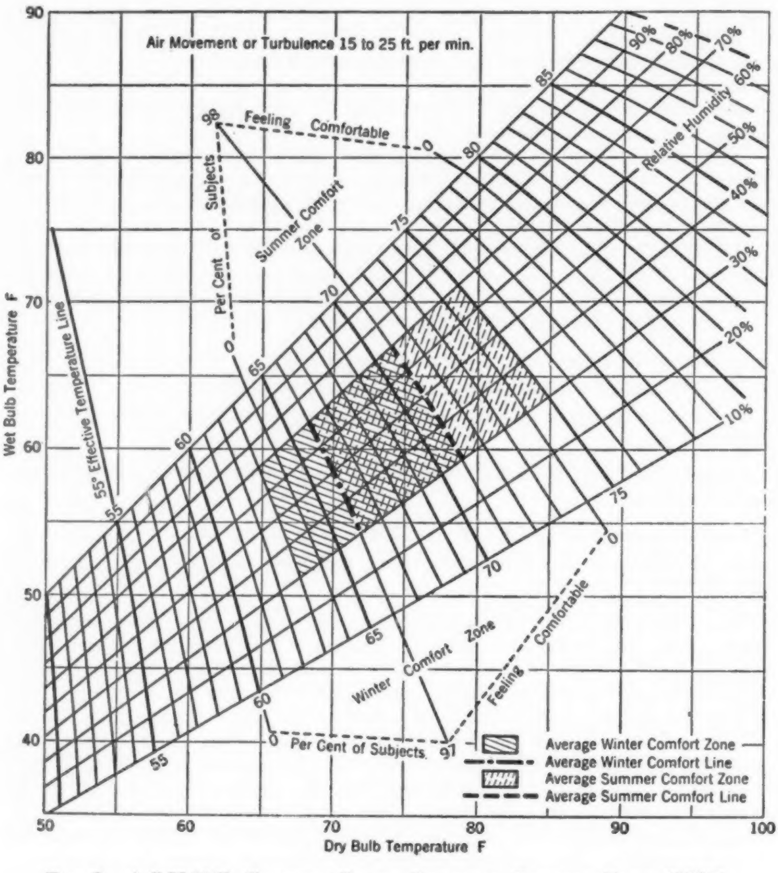


FIG. 2. A.S.H.V.E. COMFORT CHART SHOWING COMFORT ZONE (1932).

In order to have a simpler form of presentation, the *discomfort* chart was prepared by the author in 1938 (Fig. 3). Plot of the winter data immediately indicated a very unlikely form of graph for statistical data on physiological response, as it would not be expected that a large number of subjects would go precipitously from comfort to cold on a drop of one or two degrees. A graph

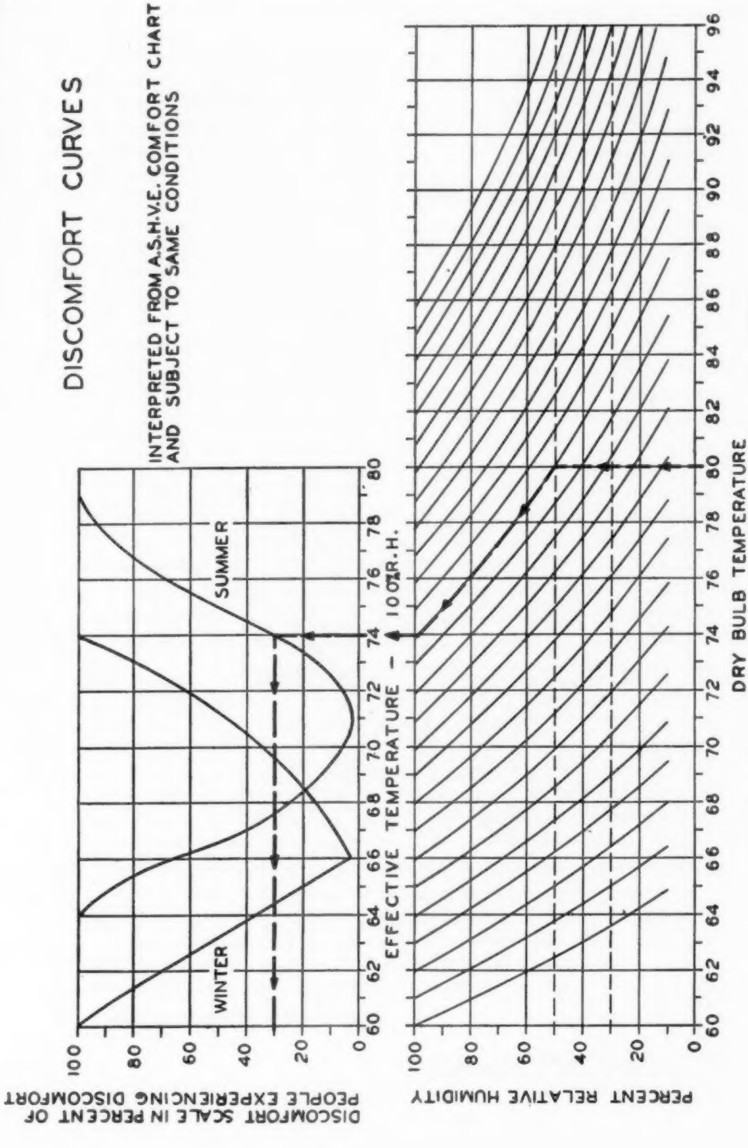


FIG. 3. DISCOMFORT CHART PREPARED BY AUTHOR (1938).

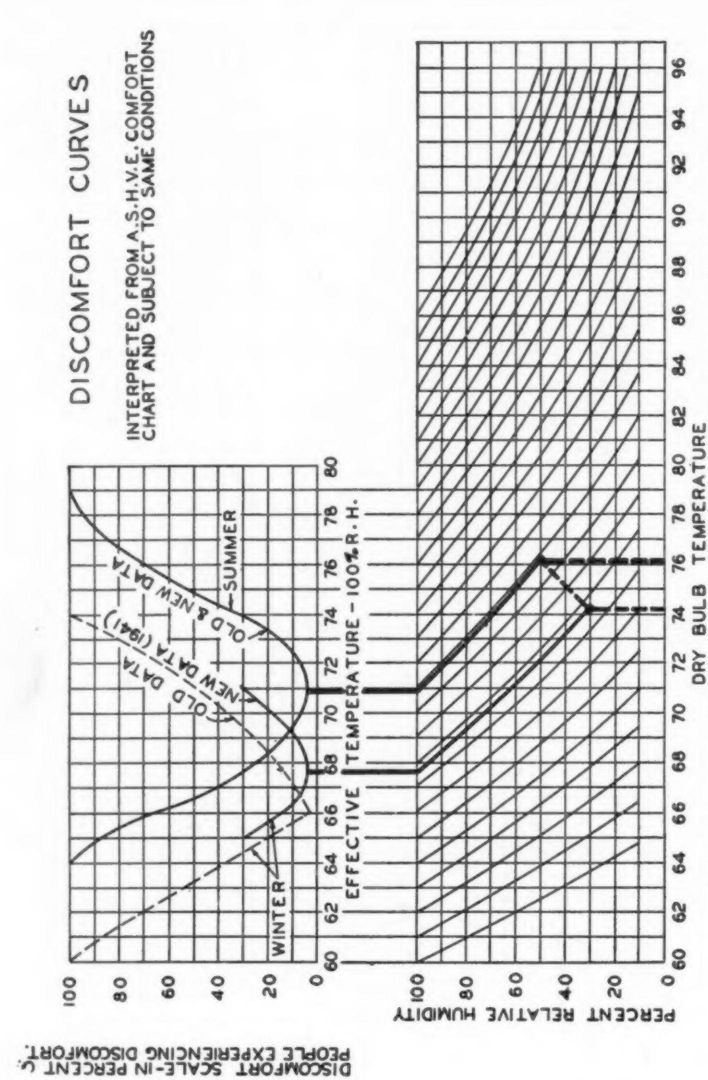


FIG. 4. REVISED DISCOMFORT CHART PREPARED BY AUTHOR (1943).
(BASED ON A.S.H.V.E. RESEARCH—1941)

such as that shown for summer is more logical, as this graph closely approximates a probability curve.

This questionable curve could result from experimental technique or from an attempt of the subjects themselves to interpret the data. Fortunately, experiments [1] conducted by the same laboratory in 1941 showed that the optimum effective temperature was approximately 67.5 ET (Effective Temperature), instead of the 66 ET of earlier experiments. At 67.5 ET, this would mean a winter temperature at 30 percent RH (relative humidity) of 74 F, instead of 72 F shown on comfort charts, and is in substantial accord with the field observations previously stated. In actual installations, temperatures tend toward the high side, as people who are too cool appear more annoyed than people too warm. If an attempt is made to account for the shift in desirable winter effective temperature by a mass change due to acclimatization, we are still confronted with our inability to explain the improbable shape of the original curve, other than by belief in the virture of enduring cold indoors.

Some experimenters have used the words *pleasant* and *unpleasant* to denote the opinion of the test subjects. It is frequently pleasant to be stimulated or calmed, but neither condition is necessary to optimum comfort of a group.

The revised discomfort chart (Fig. 4), prepared by the author in 1943, indicates the new winter data. Only a portion of the winter line is shown, as data for the complete line are lacking.

The comfort chart (Fig. 5) from the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1946 has added curves showing the distribution of comfort, but still retains the comfort zone with limits of 30 to 70 percent RH, and the winter line in accordance with the 1929 data. The text, however, calls attention to the fact that the 1929 winter data are not supported by later tests, and that the 30 to 70 percent limit on relative humidity has inadequate experimental support.

Fortunately, the substantial agreement of field observation and the A.S.H.V.E. Laboratory tests provides a firm basis for design. Control is rendered relatively easy as, in many cases, throughout the year it is unnecessary to change thermostat settings. For example, a room with outside exposure will tend to drift above thermostat setting in summer and below in winter. Thus, a setting of 75 F will substantially span the 74-76.5 F required, as shown by the line on Fig. 4 connecting 74 F, 30 percent relative humidity with 76.5 F, 50 percent RH.

The foregoing discussion pertains to building occupants above 17 years of age, and is further limited to a relative humidity range of 25 to 55 percent. The vast majority of office building applications fall within these limits. The effect of relative humidity on discomfort, shown by A.S.H.V.E. tests at other than optimum conditions, is neither questioned nor affirmed.

Experiments [2] (1941) show that a change of about 6 deg ET (approximately 7.5 deg DB) is required for one individual to go from *comfortably cool* to *comfortably warm,* or vice versa. Why, then, the need for such accurate control? First, one individual cannot be warm or cool if he wishes to remain unaware of atmospheric and temperature conditions, as the slightest change in

---

[1]A.S.H.V.E. RESEARCH REPORT No. 1172—Radiation as a Factor in the Sensation of Warmth, by F. C. Houghten, S. P. Gunst and J. Suciu, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 93.)
[2]Loc. Cit. See Note 1.

air motion or temperature would be noticeable; and secondly, we are usually dealing with more than one person.

Assume that for five subjects the extreme range of temperature is 70, 71, 72, 73, and 74, respectively, in each case, plus 7.5 deg. Graphing these results, as

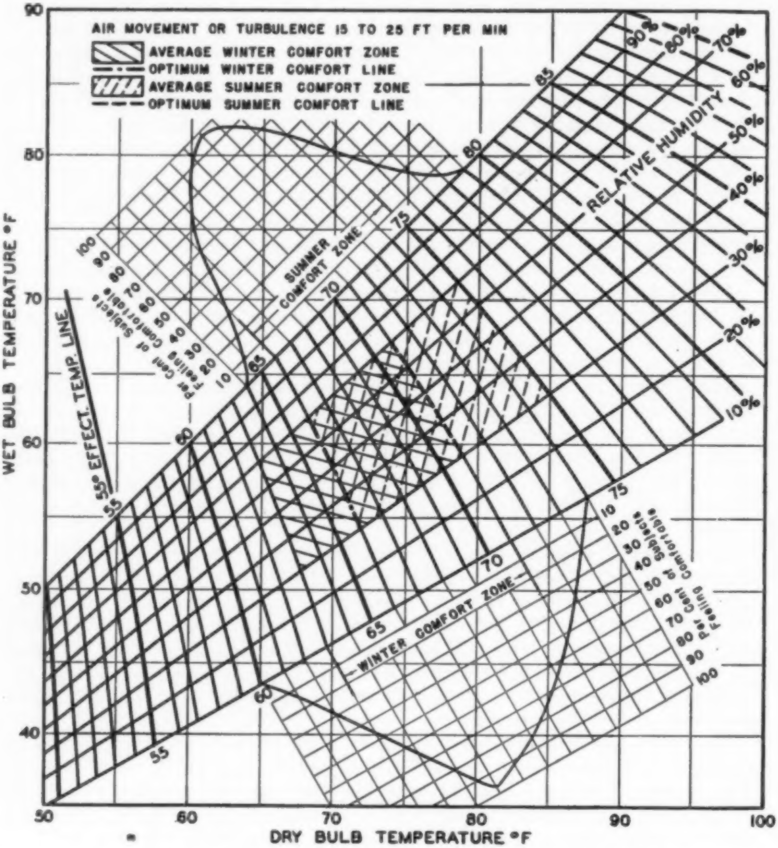


FIG. 5. COMFORT CHART FROM HEATING, VENTILATING, AIR CONDITIONING GUIDE 1946.

in Fig. 6, shows that the range 74-77.5 F lies within the theoretical tolerance for all subjects, but that for safe design the acceptable band would be 74.5 to 77 F. The figures are not intended to represent a known distribution of temperature tolerance, but only to illustrate how a reasonable tolerance of an individual will still necessitate a very close tolerance for a group.

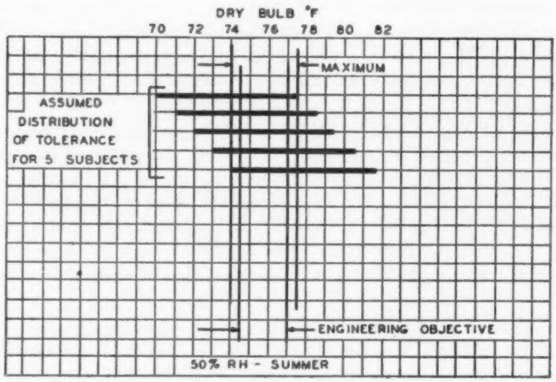


FIG. 6. ILLUSTRATION OF RELATION OF GROUP TO INDIVIDUAL TOLERANCE.

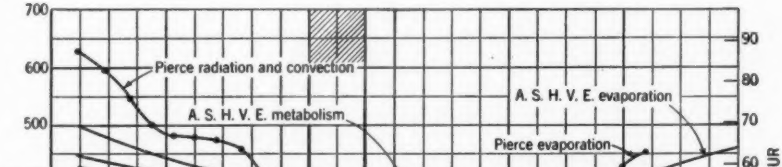


FIG. 7. RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION AND OPERATIVE TEMPERATURE FOR THE CLOTHED SUBJECT.

Are the optimum conditions herein outlined good or bad, from the health angle? No published data of the author's admittedly limited knowledge cast any reasonable doubt as to their being satisfactory, especially as the human animal is adapted to and, on a normal day, is exposed to considerable variation in temperature.

An inspection of Fig. 7 from THE GUIDE 1946, to which has been added the recommended temperature range of 74.5 to 77 F, fails to indicate any danger point. For the lightly clothed semi-reclining subjects of the Pierce Laboratory [3] experiment, the point of no heat storage occurs at approximately 77.5 F, and for the normally clothed subject of the A.S.H.V.E. tests, there is a considerable range of no storage (the difference between lines which show metabolism and total heat loss). The selected limits fall in a range of satisfactory evaporative regulation.

The conclusion would appear to be that the designated limits are approximately right for comfort, and there would be no reason to deviate unless, and until, it could be demonstrated that people should be made uncomfortable for the good of their health. The burden of proof is on the advocates of low winter and high summer indoor temperatures.

Buildings in general, office buildings in particular, are conditioned in order that the occupants may work without the discomforts due to temperature and atmosphere. Conditioning such a space provides more pleasant living and, as such, is one of many considerations in the complex employer-employee relationship. This phase may be the major consideration.

In an office building with adequate windows, in the absence of air conditioning, employees tend to blame nature for their discomforts, but the moment it is air conditioned and the windows are closed, management assumes responsibility. With an adequate conditioning system they should reach their objectives of good employee-employer relationship; with an inadequate system they may not only fail to gain good will but may actually incur ill will. There would appear to be little justification for an installation which appreciably compromises with optimum results as to comfort conditions.

DISCUSSION

F. C. WOOD, York, Pa. (WRITTEN): The author, in my opinion, makes sense, as to both the general theme and the conclusions he has drawn.

The striving for improvement in quality, stressed today possibly more than at any time in the past, is a highly commendable trend in the air conditioning art. Certainly this search for better air conditioning should not overlook what is probably the major factor affecting optimum comfort, or minimum discomfort—air temperature. I agree with the author that design should seek to produce the optimum in comfort, and am of the opinion that today there appears to be little justification, economically or otherwise, for appreciable deviation from this objective.

The willingness in the past to accept the *80 F and 50 percent*, and warmer summer cooling design conditions, as comfort criteria has doubtless been motivated largely by the desire for more economical comfort conditions than are obtainable at the optimum, as established by A.S.H.V.E. tests and indicated on the Comfort Chart. Also, there

[3]A.S.H.V.E. RESEARCH REPORT No. 1107—Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice, by C.-E. A. Winslow, Thomas Bedford, E. F. DuBois, R. W. Keeton, André Missenard, R. R. Sayers and Cyril Tasker (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 111.)

has probably been the belief that *80 F and 50 percent,* or warmer, was *cool enough,* and represented desirable comfort conditions.

Field practice often has not substantiated the latter contention. Many installations, particularly in office buildings, designed for these warmer conditions, were ultimately operated at, or close to, the A.S.H.V.E. optimum with the thermostats set to produce conditions of 76 to 78 F or lower. This seemed to insure the least number of complaints of an *uncomfortably warm* origin, from the occupants, leading to the conclusion that the reactions of a large group of people, uninhibited by the knowledge that they were being tested and having the further advantage of numbers, nevertheless agreed with the A.S.H.V.E. Laboratory tests. As the author brings out, such agreement between field reactions and laboratory tests provides a firm basis for design.

The degree of success obtained in maintaining these lower-than-design temperatures continuously in the office buildings referred to, depended on such factors as available excess plant capacity, newness and condition of the equipment, building thermal lag, duration of outside peak temperatures, actual operating load factors, etc.

Therefore, it would seem to be good logic to design installations in anticipation of ultimate usage demands as to temperature conditions. The premium in additional cost to produce and maintain the optimum comfort temperatures should be small in terms of percentage, and there is ample evidence today that the public will pay for improvements of this type that reflect better quality in air conditioning. The attention and consideration being given to air cleaning and purification, odor elimination, radiant heating, better zoning and selective individual space temperature control, and other refinements substantiate this assumption and further support the author's conclusion that *there would appear to be little justification for an installation which appreciably compromises with optimum results as to comfort conditions.*

W. A. GRANT, Fayetteville, N. Y. (WRITTEN): Our industry has long needed the practical approach to the problem of optimum room conditions, which the author has so ably presented. There is no substitute for field proof of every kind of laboratory test, and in this instance the proof is in the *conditions that the occupants are using* day after day.

Many of us who have made observations over the years are completely non-plussed that so many people, including engineers and men of science, still labor under the delusion that 68 to 70 F room temperature is comfortable in the winter time. The cold facts of *what temperature people use* simply do not bear out this preconception. While there is a band of acclimatization between summer and winter, it appears to be much smaller than the A.S.H.V.E. data have led people to suppose, and it is time that the winter data in particular be most carefully reviewed. It may be that the newer 1941 data are more nearly correct for the northern part of this country, but my own observations indicate that they are definitely on the low side for the milder areas.

For summer design conditions, there seems to be less and less justification for continuing the commercial standard of *80 F and 50 percent* as a basis of design for systems with continuous occupancy. It would be much more rational to design for a condition within the band that the author has proposed, such as 76 or 77 F, which can be maintained during the normal peak summer weather. The slight penalty in additional first cost is unimportant, if we take the position that the purpose of summer air conditioning is to make people comfortable rather than less uncomfortable.

This paper again raises the question *what is the importance of relative humidity in relation to comfort and health?* Outside of including it as one of the factors in the effective temperature index, the Society has shed no light on this important question, for either summer or winter. Practising engineers do not yet have any substantiation that winter humidification is a good thing, or that low humidities in summer are preferable to high humidities. Considering all the research that has been done on comfort, this is an embarrassing position to be in, and an organized attack on the problem is badly needed.

I think the Society is greatly indebted to Mr. Leopold for his clear-thinking, common-sense paper.

JOHN EVERETTS, JR., San Francisco, Calif. (WRITTEN): The author has, as usual, attacked the problem from the right angle instead of the wrong angle.

For many years we have been trying to maintain and meet an ideal condition of comfort which, of course, has never been reached. The optimum condition of comfort

is based upon an evaluation of the opinions of people who are asked *Are you comfortable?* This immediately raises the doubt in their minds as to whether they are or are not comfortable . . . they may be either a little warm or a little cool. However, if the same people were asked *do you feel any discomfort?* then the question would be from another angle—as to whether they may feel a little warm or a little cool but *not* necessarily experience discomfort.

I definitely feel that the comfort chart as set up by the Society is an excellent means of obtaining or attempting to obtain an ideal condition, and should be used in all design work for comfort cooling. I think that the author might go a step farther on his discomfort curves and, instead of correlating the present comfort chart into a discomfort curve, actually determine, or have determined from observations and public opinion, exactly where the discomfort lines may appear. I am sure that we would find quite a divergence in thinking if the term *discomfort* were applied in the place of the term *comfort*.

My highest compliments to the author on his usual logical thought and his clear presentation of this subject.

1319

No. 1319

A METHOD FOR IMPROVING THE EFFECTIVE TEMPERATURE INDEX

By C. P. Yaglou*, Boston, Mass.

SINCE 1923, when the effective temperature (ET) was developed[1,2] many laboratory and field workers have found that this index overestimates the influence of humidity on sensations of warmth and comfort at ordinary temperatures, and underestimates the effect in very high temperatures which approach the limit of human tolerance to heat. The latest study by Rowley, Jordan and Snyder[3] adds to the evidence, and emphasizes the need for additional work that will increase the usefulness of ET to problems in comfort cooling and heating.

The writer, who took part in developing the ET, believes that the trouble is due to inadequate experimental technique, and proposes a rational method for overcoming the disability in the light of newly developed principles, which are discussed in this paper.

Reappraisal of Technique of Determining the ET Index in the Light of Present Knowledge

Effective temperature was derived from *instanteous* thermal impressions of subjects while they were passing back and forth from one conditioned room to another. In establishing combinations of temperature and humidity which felt equally warm, one of the rooms was kept at a high humidity, with relatively low temperature, while the other room was adjusted to a lower humidity, but higher temperature, so as to make the two rooms feel alike in warmth. Two trained subjects recorded their sensory impressions (cooler, warmer, or no difference) immediately upon changing back and forth from one room to the other, without allowing time for physical or physiological adjustments. This was a matter of necessity rather than of choice, because thermal sensitivity diminished rapidly, and after a short period the observers were unable to perceive large differences of humidity or temperature differences of less than 2 to 3

*Professor, Industrial Hygiene, Harvard School of Public Health. Member of A.S.H.V.E.

[1]Exponent Numerals refer to References.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Coronado, Calif., June 1947.

deg between the two rooms, as contrasted with a sensitivity of 0.5 deg which it was possible to obtain immediately upon changing from one room to the other. Judgment of warmth depended largely on sensations of the face and hands of clothed subjects, and of the torso in subjects stripped to the waist. This was particularly true in experiments with high air velocities. The humidity effect obtained by this technique in ordinary temperatures was inordinately great, and could not be verified by other investigators using different methods of approach.

Overestimation of humidity effects, in the absence of sweat gland activity, can now be explained largely by adsorption and desorption phenomena and by failure to take adaptation (*acclimatization*) into account. When the subjects passed from a relatively dry atmosphere to a moist and cooler one, the exposed

TABLE 1—MEAN SKIN TEMPERATURE UNDER COMFORTABLE AIR CONDITIONS WITH VARIOUS AMOUNTS OF CLOTHING

(Subjects at rest. Air Movement 20-30 fpm)

	Clothing Worn (In Season)	Clothing Weight Excluding Shoes Lb	Comfortable Air Conditions			Mean Body Temp		Number of Persons
			F	Relative Humidity %	ET	Skin F	Clothing F	
MEN	Shorts only	0.1	82	54	76	92.7	......	6
	Summer clothes	3.0	76	52	71	93.2	86.1	17
	Winter indoor clothes	6.4	71	24	65	92.3	82.6	18
	Winter outdoor clothes	14.5	55	45	54	91.8	67.6	2
	Arctic pile clothing	18.3	30	78	30	90.7	......	3
WOMEN	Shorts and brassiere	0.2	83	48	76	92.3	92.1	7
	Summer clothes	0.7	80	55	74	92.7	90.3	23
	Winter indoor clothes	1.5	76	34	69	91.9	87.5	23

skin and clothing adsorbed moisture and the heat of adsorption imparted a transient sense of warmth. Upon their return to the drier room, excess moisture quickly evaporated, producing a transient cooling effect. The investigators, being unaware of these adsorption phenomena, had attributed the rapidly diminishing humidity effect largely to adaptation, and had proceeded on the assumption that all combinations of temperature and humidity which felt alike in warmth by first impression would probably result in the same degree of adaptation after prolonged exposure.

This assumption is now known to be false. It is generally recognized that reactions of individuals on exposure to any given air condition are considerably modified by previous thermal experience, and by the initial condition of the skin and clothing in regard to moisture.

Attempts to verify the seeming inordinate humidity effect by exposing subjects to more or less comfortable temperatures with different humidities were unsuccessful. In continuous exposures of 3 to 4 hours, the subjects readily adapted themselves to the test conditions and showed no significant difference in rectal

temperature, pulse rate, blood pressure, moisture loss, or impressions of comfort with different humidities.

On the other hand, in temperatures above 90 F, when the subjects began to sweat, humidity became increasingly important in affecting the rate of evaporation of sweat and sensations of comfort and discomfort. When the external temperature approached or exceeded that of the body's surface, humidity became the most important of all environmental factors. Physiological reactions in this high temperature field closely followed the ET index[4].

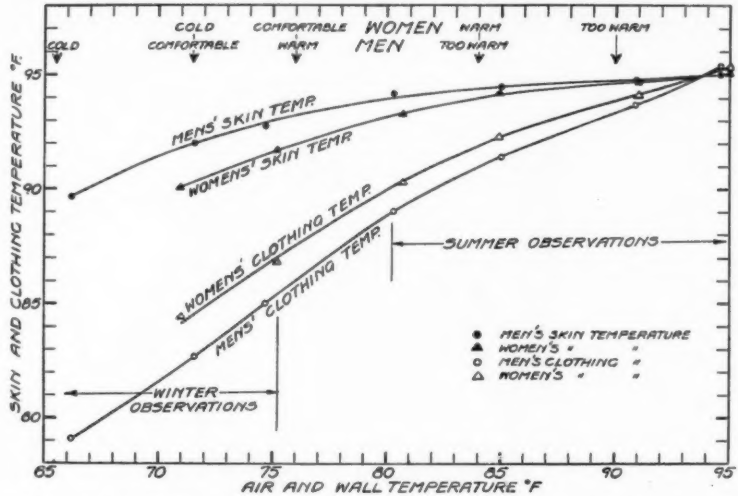


FIG. 1. NORMAL MEAN SKIN AND CLOTHING TEMPERATURES IN RELATION TO ENVIRONMENTAL TEMPERATURE. HUMIDITY NATURAL; AIR MOVEMENT 20-30 FPM.

The present status of effective temperature is therefore questionable from the standpoint of humidity and humidity control for comfort heating or cooling. Its real value is in the warm and humid atmospheres, where adsorption effects are insignificant because the skin becomes moist with perspiration. No other index can compete with effective temperature in that region, except when radiation effects are important.

MEAN SKIN TEMPERATURE AS AN INDEX OF WARMTH

Although recent studies in body heat regulation have failed to develop a more practical thermal index than ET, they have contributed much fundamental information which can be utilized for improving the ET. One of the most significant and consistent findings, notably by Hardy and DuBois[5], and by Gagge, et al[6], is the existence of a close relationship between mean skin temperature and

comfort in air conditions that lie below the zone of evaporative cooling, *i.e.*, when persons are not sweating.

Yaglou and Messer [7] have verified the validity of skin temperature measurements over a wide range of practical conditions summarized in Table 1. It can be seen that men and women were comfortable, in winter or summer, when their mean skin temperature averaged between 91.0 and 93.0 F, regardless of environmental temperature (30 to 82 F), when they were suitably dressed in clothing which weighed from a minimum of 0.1 lb at a temperature of 82 F to a maximum of 18.3 lb at 30 F, excluding weight of shoes.

The normal mean skin and clothing surface temperatures, under ordinary winter and summer conditions, are shown in Fig. 1. Barring emotional dis-

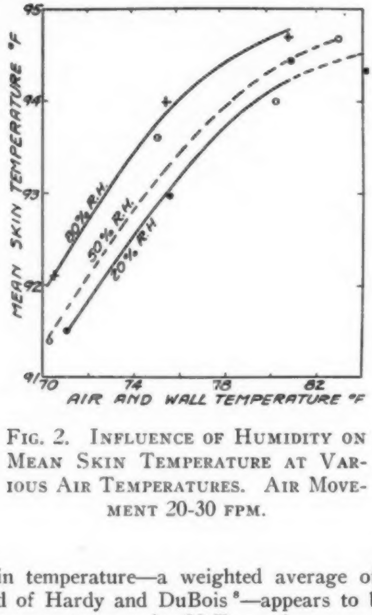


FIG. 2. INFLUENCE OF HUMIDITY ON MEAN SKIN TEMPERATURE AT VARIOUS AIR TEMPERATURES. AIR MOVEMENT 20-30 FPM.

turbances, mean skin temperature—a weighted average of 17 readings computed by the method of Hardy and DuBois [8]—appears to be a fairly sensitive index of warmth in temperatures under 82 F, or when persons are not sweating. When perspiration sets in, the skin temperature curve begins to flatten out (see Fig. 1), owing to cooling by evaporation of sweat. Beyond this point, skin temperature rises very slowly and cannot, therefore, be taken as an accurate index of the degree of warmth or of discomfort experienced.

Skin temperature also fails as an index of warmth on exposure to sudden temperature changes while the skin temperature is rapidly changing, or during heavy muscular work, when the skin temperature usually drops, owing to diversion of blood to the active muscles.

Despite these limitations, mean skin temperature is the best objective index now available for studying the effects of thermal factors affecting comfort in

resting persons. In fact, the skin constitutes the primary thermostat of the body. Its temperature reflects the effects of many factors, physical and physiological. On the interior it is affected by metabolic rate, blood circulation, nervous reflexes, body posture, etc., while on the exterior it is affected by clothing, air temperature, humidity, air movement, radiation, etc. Any factor affecting the rate of heat loss from the skin exerts its influence through a change of skin temperature.

HOW TO CORRECT THE ET INDEX

It is evident from the foregoing that if ET were an accurate index of warmth it would closely follow the mean skin temperature; in other words, the ET

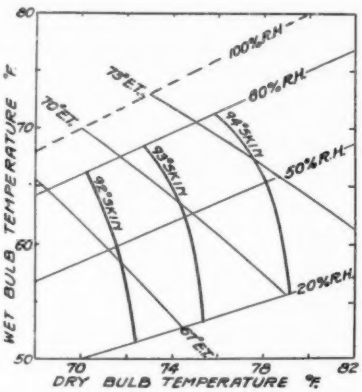


FIG. 3. LINES OF EQUAL MEAN SKIN TEMPERATURE IN RELATION TO ET AND TO DRY AND WET BULB TEMPERATURES. AIR MOVEMENT 20-30 FPM.

lines, as drawn on a psychrometric chart, would also be lines of equal mean skin temperature, at least under thermal conditions which induce no sensible perspiration. On the other hand, should the humidity have no effect on warmth, the lines of equal skin temperature should run parallel to the dry bulb lines. Any departure of constant skin temperature lines from those of the dry bulb would be due to humidity, if all other factors were kept unaltered.

An exploratory study was made in a conditioned room, to measure skin temperatures at approximately 20, 50 and 80 percent RH, with temperatures between 70 F and 85 F. Two young men were exposed to these conditions for three hours before final observations were made. The men wore customary indoor winter clothing, weighing on the average 5.3 lb, excluding weight of shoes. Skin temperatures were measured by thermocouples on 17 areas under the clothing, and by a radiation thermopile on 6 exposed areas of the head and

hands. Mean skin temperatures were computed by weighting the readings with DuBois' area factors [5]. The results are shown in Figs. 2 and 3.

With reference to Fig. 3, humidity appears to affect skin temperature measurably, but the effect is less than half of that indicated by the ET index. For instance, at 70 deg ET, an increase of humidity from 20 to 80 percent is equivalent to a decrease of nearly 3 F deg in air temperature on the basis of skin temperature, as against a decrease of 7 F deg indicated by ET.

Influence of humidities of less than 50 percent on skin temperature appears to be negligible; the subjects themselves were unconscious of any definite humidity sensations in this region after the entry contrast wore off. With 80 percent relative humidity, on the other hand, there was no difficulty in perceiving the high humidity and, when questioned, the subjects expressed preference for a lower humidity, although the objection to the higher was not great.

On the strength of this exploratory evidence it seems feasible to correct the ET index for humidity, and probably for radiation also, by substituting lines of constant skin temperature for the present ET lines, as shown in Fig 3. Final results should be established by a more thorough study, using a greater number of subjects and covering a much wider range of temperature. It is important that subjects, as well as technique of measuring skin temperatures, once standardized, be kept unchanged throughout the study.

SUMMARY

1. Effective temperature is shown to overestimate greatly the influence of humidity in the range of temperatures met with in comfort heating and cooling.

2. The disability is attributed to adsorption and adaptation phenomena, which had been overlooked in the development of this index.

3. A rational method is proposed for correcting the index on the basis of mean skin temperature, which is now known to be the best objective index of comfort.

4. It is recommended that the Society's Research Laboratory make a comprehensive study of the problem with a view toward settling once and for all, this important limitation of effective temperature.

REFERENCES

1. Determining Lines of Equal *Comfort,* by F. C. Houghten and C. P. Yaglou. (A.S.H.V.E. TRANSACTIONS, *29*:163, 1923.)

2. Effective Temperature with Clothing, by C. P. Yaglou and W. E. Miller. (Ibid., *31*:89, 1925.)

3. Comfort Reactions of 275 Workers During Occupancy of Air Conditioned Spaces, by F. B. Rowley, R. C. Jordan and W. E. Snyder. (See Chapter No. 1321.)

4. A.S.H.V.E. RESEARCH REPORT No. 654—Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten. (A.S.H.V.E. TRANSACTIONS, *29*:129, 1923.)

5. Differences Between Men and Women in Their Response to Heat and Cold, by J. D. Hardy and E. F. DuBois. (*Proceedings National Academy of Sciences, 26*:389, June 1940.)

6. The Influence of Clothing on the Physiological Reactions of the Human Body to Varying Environmental Temperatures, by A. P. Gagge, C.-E. A. Winslow and L. P. Herrington. (*American Journal of Physiology, 124*:30, Oct. 1938.)

7. The Importance of Clothing in Air Conditioning, by C. P. Yaglou and Anne Messer. (*Journal of the American Medical Association, 117*:1261, Oct. 11, 1941.)

8. The Technique of Measuring Radiation and Convection, by J. D. Hardy and E. F. DuBois. (*Journal of Nutrition, 15*:461, May, 1938.)

DISCUSSION

C.-E. A. WINSLOW, New Haven, Conn. (WRITTEN): The Society is to be congratulated on the admirable paper by Professor Yaglou. Many of us have been conscious for some time, of the fact that the present Comfort Chart does not reflect the true effects of humidity in the Comfort Zone. It is also quite clear, as Professor Yaglou has pointed out, that in this area skin temperature is the best measure of comfort. By a simple and direct experimental procedure he has, in this brief contribution, cut right into the heart of the problem and effectively solved it.

The investigators at the Society's Research Laboratory, in the early days in Pittsburgh, did a magnificent piece of work, but the progress of physiological knowledge since that time has now given us far more effective approaches to the problem. It is not a reproach to the pioneers but a tribute to their vision that, in the light of new knowledge, their work should be revised and brought up to date.

JAMES D. HARDY, M. D., New York, N. Y. (WRITTEN): Professor Yaglou's proposal to use the average skin temperature as an objective index of comfort represents a real step forward in the rational treatment of the important practical problem of comfort. It is also of importance that Professor Yaglou has included women in his studies. It might be desirable to include older people (age 60 years or more), as they will probably constitute an increasing proportion of our population to whom air conditioning will be important. I should like to point out in this connection that, in our studies on nude men and women in the Russell Sage calorimeter, the average skin temperatures of the two sexes was quite different under the same circumstances, the women having lower skin temperatures in the cold and higher skin temperatures in the warm environments. Therefore, some care will be necessary in evaluating skin temperature values.

Comfort is essentially a sensation problem, and thermal comfort probably represents an environmental exchange in which the number of afferent impulses arising in the skin is minimal. As these impulses are due to changes in skin temperature about a mean value, the mean value of skin temperature, although closely related to comfort, is probably not the basic factor. For example, in uncomfortably warm environments, it is probably the rise and fall in skin temperature, due to periodic evaporation and production of sweat, which gives rise to the thermal sensations, and the same sensations are probably important in the regulation of body temperature in such environments.

Also in the zone of body cooling, periodic shivering is due partly to the sensation of cold arising from a falling skin temperature. The extreme sensitiveness of the skin to changes in temperature is recalled in this regard: threshold for warmth, a rise of 0.0008 centigrade degree per second for three seconds; threshold for cold, a fall of 0.004 centigrade degree per second for two seconds. I suggest that the average value of the wetted area of the skin may serve as a useful index in the zone of evaporative cooling when the average skin temperature changes but little.

I concur heartily with Professor Yaglou's suggestion that radiation should be included, under some circumstances, as an important factor. The question of variable forced air currents and their effects on the skin temperature and evaporation losses is another important environmental factor.

W. J. MCCONNELL, M.D.,[ΔΔ] New York, N. Y. (WRITTEN): Professor Yaglou has proposed a rational method, verified by significant experimental evidence, for improving the usefulness of the ET Index in comfort cooling and heating which merits

[ΔΔ]Assistant Medical Director & Director, Industrial Health Section, Metropolitan Life Insurance Co., New York, N. Y.

favorable consideration. It is expected that the study outlined will furnish data for correcting the effective temperature lines for humidity effects within the range of ordinary temperatures, and thereby greatly enhance the value of the ET Index.

THOMAS CHESTER, Detroit, Mich. (WRITTEN): This valuable paper convincingly indicates the need for improving the ET Index, and what is more important, how to do it. Research should be started forthwith, in order to eradicate long recognized defects.

In explaining why humidity effects were originally overestimated the statement is made that "in passing from a relatively dry atmosphere to a moist and cooler one, *in the absence of sweat gland activity,* the exposed skin and clothing adsorbed moisture and the heat of adsorption imparted a transient sense of warmth." This supposition of the entire absence of perspiration with skin temperatures under the clothing, of around 90 F does not seem to be warranted. When the skin perspires, it does so with a controlled and graduated action, instead of a quick on and off effect. The formation of small drops is first noticeable on the forehead and palms of the hands, because of the more numerous sweat glands per unit of area at these places. When such incipient exudation is visible, it is conceivable that evaporation is occurring from the pores of the skin beneath the clothing. On this basis it would be simpler, and perhaps more accurate to attribute variations in sensations of warmth to variations in evaporation, instead of to adsorption or desorption. Evaporation is governed chiefly by differences in vapor pressure. If there is any evaporation of perspiration, a subject, in passing from a relatively dry atmosphere to a moist and cooler one, will for a short time lose less latent heat because of arrested or diminished evaporation, if the vapor pressure of the water vapor in the air in the new environment is higher than in the former one. This reduction in the rate of heat loss from the skin is felt as an increase in warmth. The author goes on to state that *upon returning to the drier room, excess moisture quickly evaporated, producing a transient cooling effect.* This, of course, is correct and is due to the pronounced tendency of moist substances to fall to the prevailing wet bulb temperature. As regards the adsorption theory it is very questionable whether any instrument would show any increase in the temperature of clothing, removed and apart from a human body, due to moisture regain and the concomitant release of heat caused by the condensation of water vapor within sub-microscopic pores of the fabrics. Regardless of conjectures of how skin warming and cooling sensations are produced, they very definitely are produced when variations occur in the rate of evaporation of perspiration, and divergencies between effective temperatures and average skin temperatures are to be expected.

In carrying out research work on this subject, the usual complexities are to be expected. Perhaps it would be difficult and expensive to have the subjects similarly clothed. If this could be done, it would at least eliminate one variable of considerable influence.

The author undoubtedly knows whether or not there is much variation in normal skin temperatures of different individuals when similarly attired, if subjected to the same atmospheric conditions, and a statement from him on this matter would be appreciated. If there is much variation it could, of course, be suitably dealt with in the production of informative data.

DOUGLAS H. K. LEE,* Brisbane, Australia, (WRITTEN): I will discuss Professor Yaglou's paper under three headings: (1) Inadequacies in present ET (effective temperature) scheme; (2) Expected nature of modifications; and (3) Suggestions.

1. ***Inadequacies in present ET scheme*** are indicated by:

a. Overestimation of the heating effect of increased humidity at intermediate temperatures.

b. Non-recognition of the *cooling* effect of high humidities at low temperatures, especially with clothing.

c. Non-inclusion of radiation factor.

Note: Data obtained in my laboratory on the relative effect of humidity at high temperatures are now under examination. I should not like to commit myself on the matter at the moment.

---

*Physiology Department, University of Queensland.

2. *Expected nature of modifications* are:

a. Thermodynamically, when the body is not actively sweating and the only moisture available for evaporation is that of percutaneous transudation, the rate of which has been shown to be relatively constant, the only effect brought about by an increase in absolute humidity should be that arising from diminished evaporation from the respiratory tract. The extent of this transfer of heat loss from pulmonary evaporation to cutaneous radiation-conduction-convection can be determined from psychrometric tables. The amount would be small and the effect on comfort should be equally small. The environmental temperature fall which would be necessary to balance this could probably be estimated from the John B. Pierce Laboratory data. I should, offhand, expect even less deviation from the dry bulb temperature lines than Professor Yaglou gives in his Fig. 3.

b. Damp air and damp clothing (after equilibration with the atmosphere) offer less thermal resistance and increase heat loss appreciably at moderately low temperatures (35-45 F). It might be expected that the comfort lines, especially for the clothed man, would cross somewhere about 45 F with this reversal of humidity effect.

c. Bedford's recommendation to use globe thermometer temperatures (uncorrected), instead of dry bulb temperatures, appears to be justified empirically where radiation is not sharply localized. Alternatively, a scale could probably be worked out for determining radiation corrections to the dry bulb scale where mean radiation temperature of the surroundings is known from thermocouple or radiometer measurements.

Where radiation is intense and from a limited source, (*e. g.* sun fire), the geometry of the globe is so different from that of man that a simple globe thermometer reading might be expected to be inadequate. Moreover, the relative importance attached by man to localized radiation in the assessment of his general comfort is not known.

A further practical difficulty in the assessment of radiation effect is introduced by the time factor. Exposed skin quickly warms up, giving immediate sharp sensation with some subsequent fading, but the heat contribution is immediately added to the total load, which probably also has some influence in sensations of comfort. Clothed skin, on the other hand, warms up quite slowly. Just what shape-curve of sensation will result from a given situation is hard to predict, depending upon time, relative areas, nature of clothing, and intensity and quality of radiation.

3. *Suggestions:* I would make a strong plea for maintaining the original criterion used in the ET scheme; that is, keep it as a comfort index. By all means, use other criteria—skin temperatures, thermodynamic predictions—as *checks,* and thoroughly investigate discrepancies to see if the cause lies in the experimental method of assessing comfort (Yaglou suggests one such case) or is inherent in the checking criterion (pulse rates are notoriously bad criteria). But make the final criterion one of comfort. This is not only logical but practicable, since comfort is the item with which the average worker is concerned.

I firmly believe that the final scientific assessment of thermal stress, and thus of potential thermal strain, will be a thermodynamic one; but the day of its realization is still far off. The ET scheme has been the best empirical and practical index to date, but it can be improved. By all means let us have it improved, but retain its basic character.

I would support Professor Yaglou's suggestion that the A.S.H.V.E. Research Laboratory make a comprehensive review of the ET scheme, and would suggest that its application to fairly low temperatures be also reexamined. If there are to be amendments, and I think there should be, let them all be made at the same time so that the resultant chart can be used for another 20 years without alteration.

Thomas Bedford, London, England (Written): It has been my privilege to see an advance copy of Professor Yaglou's valuable paper, and I should like to offer him my congratulations on it. I am glad to learn that the question of revising the ET scale is being discussed. I hope that the project will go forward, and I would urge that in this work radiation be taken into account.

I am much interested in the proposal that ET lines shall be drawn as lines of equal mean skin temperature. There is much to be said for using some objective index instead of comfort votes, in constructing a scale of warmth, and I agree that, until perspiration

comes into play, skin temperature is the best available index. At higher temperatures, however, I doubt whether skin temperature alone will be adequate. Then, I think, due allowance should be made for the rate of sweating, and perhaps also for the effects of heat on the heart rate and on body temperature. It may be found necessary to utilize some such overall index of physiological effect as was used by Prof. Sid Robinson and his colleagues.†

Although, for moderate temperatures, skin temperature is the best objective index of comfort, and although it may be of great value in constructing a scale of warmth, it should be borne in mind that when different persons have the same skin temperature they are not necessarily equally comfortable.

The subjective feelings of warmth of a single person can be predicted with considerable accuracy from a knowledge of his skin temperature. Thus, Miss Ward** found very high correlations between comfort votes and the skin temperature measured on the forehead or over the carotid artery. Yet, in my own observations on a large number of persons, doing very light industrial work‡ in which the skin temperatures of the forehead, hands, and feet, were measured, it appeared that skin temperature measurements were of little practical value in predicting the comfort of individual persons. The probable range of errors of estimation was such, that, when the combination of forehead, hand and foot temperatures caused the average person to feel exactly comfortable, some persons felt too cool and others too warm. On the whole, the comfort of the individual occupants of a room could be predicted rather better by estimating the average comfort vote from a knowledge of the equivalent temperature (or of the ET) of the room than by estimating the feeling of warmth of each occupant from a knowledge of his skin temperature.

It may be that by employing the mean skin temperature, as used by Professor Yaglou, a better estimate of comfort will be obtained, but that is a matter which can be tested. The results of my observations serve to reinforce Professor Yaglou's caution that the experimental subjects should be kept unchanged throughout the study.

F. C. McINTOSH, Pittsburgh, Pa. (WRITTEN): The information given in Figs. 2 and 3 might indicate a much greater relative humidity effect between 50 and 80 percent than between 20 and 50 percent. As Professor Yaglou indicates by his points, and states in his text, this information comes from a few exploratory tests. Perhaps further information would change this relationship.

Since Professor Yaglou has apparently discovered the reason for the misinterpretation of data taken 25 years ago, I would like to ask him what reason he gives for the effect of relative humidity on skin temperature; also, if he thinks the effect would be directly proportional, or otherwise as Fig. 3 would indicate; and in the latter case, if the curvature of the skin temperature lines would continue up to 100 percent?

W. L. FLEISHER, New York, N. Y. (WRITTEN): In connection with Professor Yaglou's suggestion for a method of improving the ET Index, I would take the most profound exception—not only to his conclusions—but to his obvious misconception of the whole subject.

There is really only a very narrow range of ET which are of particular importance, so far as the practical application of air conditioning to human beings in a conditioned environment is concerned. Within this very definitely determined range, practically the whole science of comfort conditioning has been built, and the importance of our art and our research is predicated upon the public acceptance of these particular environments. This range has been thoroughly substantiated in the field as correct.

Before I develop this theme further, I want to criticize Professor Yaglou's assumption that the work he did in the early days of the Laboratory is by any means the basis of, or has any important bearing upon, the results which are generally used by the public. When he says that two young men were exposed for a certain number of hours, or when he speaks of Dr. Dubois' experiments, he fails to take into consideration the summer experiments in Minneapolis, Toronto, San Antonio, Washington and

†*American Journal of Physiology*, Vol. 143, p. 21, 1945.
***American Journal of Hygiene*, Vol. 12, p. 130.
‡Industrial Health Research Board Report, No. 76, 1936.

New York—where thousands of people, for months, were subjected to very definite conditions of temperature and humidity and expressed a great preference for a narrow range of so-called effective temperatures.

Within this narrow range, I contend that the resultant combination of temperature and humidity—the humidity varying from 30 to 70 percent—combined with the temperature which created a condition along an ET line corresponding to this temperature and relative humidity, was comfortable to the vast majority of the people, and the deviation from this condition to discomfort on both sides, that is, to a cooler or a warmer condition, is so definitely indicated that there can be little possibility of improper classification.

The phase of air conditioning on which the Society has grown is a summer air conditioning function, created by the use of some type of artificial refrigeration. As I have stated, within the narrow range which I have indicated, developed by these thousands and thousands of tests, the ET is well established, and is not in any way related to the requirements of skin temperature measurements. Why the question of clothing should necessarily enter into this particular analysis, is beyond comprehension. Certainly, the 18 lb of clothing required for the Eskimos has very little to do with this subject at hand, and the reference is more than likely to create a feeling of uncertainty in the minds of the thousands, or even hundreds of thousands, of people who have benefited from the standards which were originated in the Laboratory, and conclusively proved in the field, to be correct from a practical angle.

Up to this point, I have discussed this paper purely from the angle of summer comfort, as indicated from the composite chart, Fig. 11, p. 219, in THE GUIDE, 1947. I cannot pass over this figure without again calling attention to the discovery of the Fleishers[Δ] that the effective temperature line, or so-called ET line, which, in our opinion, is the line of constant volume, passes through the 0 percent relative humidity line on the maximum comfort ET line at the normal body temperature of practically all human beings.

In this discussion of Professor Yaglou's paper, I am not going to enter into the further discussion of the kinetic energy theory developed by the Fleishers, but it has certainly, in my opinion, been much more significant than the points developed by Professor Yaglou.

It was stated earlier in this discussion, that there was another point in the so-called effective temperature setup which was of importance; and that is, outside the comfort range—in the *danger* range. If there is criticism of some of the methods pursued in the past by the Laboratory and consequently the 87 deg ET line is subject to criticism as the critical point for men at light labor, we have the work of Haldane and his associates to corroborate or confirm the work that we have done indicating that this point, or at least some point along the 87 deg ET line, is the point where some fairly serious physiological changes take place in human beings. It is my opinion that this condition is not necessarily serious, and Dr. Bedford who investigated these higher temperature and humidity conditions, in order to cover the safety of employment in the mines of South Africa, has also corroborated the findings of our own Laboratory. Of course, I am sure that Professor Yaglou will contend that these conditions were arrived at physiologically; whereas, the ET lines were determined psychologically or by sensible sensations. However, the equation of relationship between humidities either absolute or relative and dry bulb temperatures remains approximately the same for the *comfort* and *danger* zones. Of course, I must contend again that they follow, particularly in the higher ranges, much more nearly the Fleisher kinetic energy lines than they do the arbitrary lines developed by the Laboratory. Nevertheless, the relationship between temperature and humidity in the higher ranges is very similar to the equation of relationship in the comfort zones.

Actually the thing that interested me most in Professor Yaglou's paper is his casual disregard of the effect of humidity under cooler conditions. That is, the lack of any effective changes in sensations due to more or less humidity in the wintertime under cool conditions. I was not able to understand, in reading his paper, whether he was discussing this whole subject from the angle of the environmental conditions out-

[Δ]Comfort and Health and Temperature—A Mathematical Solution, by W. L. Fleisher and W. L. Fleisher, Jr. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941.)

doors or indoors. Of course, in Table 1, he mentions *summer clothes, winter indoor clothes, winter outdoor clothes* and *Arctic pile clothing,* though why he changes from *clothes* to *clothing* is more than I can understand; so that I think he has in a way confused the subject, or the subject about which we are creating discussion. In other words, he attempts to belittle work that has been very advantageous to the development of this art, and, because of his name in this art, what he says is liable to carry weight. This in my mind is not based on anything nearly so thorough or conclusive as the work done by the Laboratory over the past 25 years.

Coming back to the discussion of the field of relative humidity, or we might say, moisture content of the air in the wintertime—as he starts with 24 percent RH and goes up from that point, I think that he has completely weakened his position by not recognizing that in the wintertime we are discussing conditions inside an enclosure, such as a residence of not a 30, 40, 50 or 60 percent RH but conditions of relative humidity that may go well below 20 percent. I have discussed this question fully, many times; not only before the Society, but before various medical and health groups. On the basis that Professor Yaglou's, Dr. DuBois' and Dr. Winslow's measurements of desirable humidity in the house are all based on physiological measurements to reactions, and their contention that humidity in the winter time is relatively unimportant, is fallacious in the light of people's reactions to dry enclosures, it seems to me a mistake that the crude methods of measurements, which these experts in the line of physiology employ, should be weighed against the sensation of human beings. In most cases in winter, it is not possible to maintain more than 30 percent RH, owing to the deposit of moisture on cold surfaces. In view of conditions of relative humidity between 25 and 35 percent in the opinion of hundreds of people whom I have questioned, and thousands of people whom I have exposed to these conditions during the past few years, I am extremely critical of the continued reiteration of Professor Yaglou as to the unimportance of humidity in the wintertime.

After all, to be slightly cynical, one must not forget that Professor Yaglou was just as dogmatic about ionization and its importance as he is skeptical of the importance of humidity in the wintertime, and that his findings, backed by exhaustive tests undertaken by the Harvard School of Public Health, were completely fallacious.

In conclusion, I would like to bring out these points: The Research Laboratory of the Society, in cooperation with outside sources, has developed over a period of years completely correct zones of comfort and of physical environmental hazards. We have advocated, owing to the Laboratory's various investigations, a minimum relative humidity for comfort in the wintertime. There is a possibility that some of the Laboratory's methods were crude and inaccurate; nevertheless, empirically, they have proved practical and acceptable and the growth of the Society is due to a great extent to the usefulness and acceptance of these standards. The casual and incomplete analysis such as Professor Yaglou has given should not in any way affect the acceptance of the prior research of the Society.

A. C. WILLARD, Urbana, Ill. (WRITTEN): This paper deals with an important factor, among many, affecting human comfort in occupied spaces. It raises again the question of the true significance of relative humidity as a component of the ET Index. The objective evidence presented is sufficiently convincing to justify the serious consideration of the Society's Committee on Research.

Professor Yaglou's recommendation as shown under Summary, item 4, needs clarification as to the nature and extent of program, time, staff and funds necessary to make a comprehensive study of the problem, before final action is taken on assigning the project to the Research Laboratory to execute.

W. E. ZIEBER, York, Pa. (WRITTEN): I think the paper by Professor Yaglou is very timely and important. Several things enter into this subject which, as he points out, need further consideration, and I am satisfied the Society is the proper place to give this the thought that is required at the present time.

Professor Yaglou is in a very good position to criticize and comment on the work that has been done, and to determine the various portions of work on the subject that have been able to stand under the test of time, and those parts which have not been able to stand. Because of his past work, I think he is a better critic than most of us.

The attitude that some of the work may have been in error because of the inability to understand certain phenomena, is a very good approach. I believe we should seriously consider some rechecking, as suggested by Professor Yaglou. The conditions pointed out are certainly logical and I agree that the Society has a problem which it should study. Professor Yaglou points out that the effects of adsorption of moisture by clothing and the desorption of moisture have a heating and cooling effect. Also, the initial skin and moisture conditions, and the change under those conditions are very important. I agree with Professor Yaglou that these conditions must be carefully studied to be able to detect the proper reactions when making observations. I regret that I cannot contribute much toward the method of handling these particular phases. I believe the men who have been working on these subjects can better advise how to take care of such conditions and what type of measurements need to be taken.

Instantaneous readings may be one of the causes of difficulties and further information regarding them may result from an investigation now being made by the A.S.H.V.E. Committee on Research on the shock effect upon people when entering cooled spaces, or going from cooled spaces to warm spaces.

This phase is involved in physiological and psychological reactions of humans, but I do not see how the Society can divorce this work from the medical or public health professions. They may guide it and act as a clearing house, but the work will have to be carried on by people who are able to understand the reactions of people from these two angles.

The studies of effective temperature should provide answers to more questions than have been considered in the past; that is, we should know the relationship of shock effect and ET Index. The work should cover more than a couple of people, and it should cover people of different ages. We should know the difference between active and inactive people. The effects of velocity of air, both in cooling and heating, should be studied.

The effect of radiant heating and radiant cooling from objects, panels, and people in groups should be included in the studies because of the importance that has been put upon this particular subject in the past year or so.

Because of the magnitude of the work required to get this subject straightened out, as it appears now, I believe it is a good thing to make a correction, as proposed by Professor Yaglou, to improve the methods of selection of effective temperatures for use in laying out and designing air conditioning systems. I believe his work illustrating the relationship between skin temperature and effective temperature can be used by designers to good advantage, especially if humidity does not have much effect on sensation of warmth and comfort at ordinary temperatures. Most air conditioning is applied under ordinary temperatures, and does not approach the limits of human endurance to heat. Basically, this is the important thing to correct, if only in a temporary way at this time, considering that the problems involved are numerous and will require considerable work in the future.

If Professor Yaglou's suggestions are adopted, our comfort charts will need adjusting, since the comfort zones are constructed upon the ET lines.

I believe that any future work on effective temperature should be combined with comfort work. The two are not directly related—that is, ET lines extend beyond the comfort zones. It is possible, however, to use the same experimental technique and apparatus setup for both studies. Since the comfort zones are based upon ET lines, at present, they will need to be checked at the time the ET is worked upon to be sure of its boundaries and values.

KENNETH E. ROBINSON, Lansing, Mich. (WRITTEN): The writer is very much interested in this paper, first because, as the author states, it has been evident for some time that humidity did not vary the sensation of warmth as greatly as the ET chart would indicate, and second because of a nontechnical study the writer carried on for a period of three or four years.

This study consisted in part of checking with customers who had humidifying equipment controlled by a humidistat. Although most of these people, including physicians, had insisted on having equipment capable of keeping a relative humidity of 50-60 percent when purchasing the equipment, in every case it was found within a year's time that the humidity was generally maintained at a range of 35-40 percent and never over 40 percent.

The other part of the study was confined to a large number of customers who had humidifiers in series with a forced warm air heating system. Ordinarily it was found that this humidifier would maintain a relative humidity in the house of from 25-35 percent. However, it soon became apparent that, if the humidifier was out of service, we would get a call from the customer and the complaint was always the same: The temperature at the thermostat could be maintained, but the home was no longer as comfortable as it had been in the past. It was also found by experimentation that raising the temperature of the building one or two degrees did not compensate for the lack of humidity as concerns human comfort.

We realize that there is a vast difference between the two expressions, *sensation of comfort* and *sensation of warmth,* and it is my wish to raise the question as to whether both can be placed on one chart. I have asked many physicians what the proper humidity in a building should be in order to maintain the health and comfort of the occupants, but to date, I have not received two answers which agree.

The writer wishes to compliment Professor Yaglou on this fine paper, and sincerely trusts that the Society will see fit to carry on with the work he has outlined.

CHARLES S. LEOPOLD, Philadelphia, Pa. (WRITTEN): There can be little doubt as to the desirability of an objective test for comfort, if a dependable method can be obtained.

In the general range of optimum comfort, two statements can safely be made:

1. Practically all investigators concede that increase in relative humidity at constant dry bulb produces some increase in the feeling of warmth.

2. For the usual working range the effect of relative humidity is small.

In this paper, Professor Yaglou substantiates the general effect of relative humidity, but believes that the effect is approximately one-half as great as that previously reported. It is realized that, for the complex and adaptable structure of the human body, this is an extremely small effect to measure.

In effect, Professor Yaglou states that the present ET lines are faulty in the range of ordinary indoor comfort for normally clothed people in sedentary occupation and suggests that for temperatures 82 deg and below we determine the slope of the ET lines by establishing lines of constant skin temperature. He states that for temperatures above 90 deg, relative humidity is probably more important than previously noted and leaves a void in observation for conditions between 82 deg and 90 deg.

Consider, first, his case against the present effective temperature line.

Reference is made to the paper by Rowley, Jordan and Snyder,§ but in the light of the discussion following this paper it would appear that there is serious doubt as to the validity of the conclusions.

Professor Yaglou criticizes the original laboratory work in determining ET lines and states, without offering experimental proof, that the error is due to adsorption and desorption effects. I do not believe that the case for adsorption and desorption phenomena is as simple as it appears. Ferry Houghten pointed out in a discussion of a paper° that the dissipation of heat from a human is not equivalent to drying a dish cloth, in which case heat for evaporation must come from without, but that it is more akin to a heated cylinder with a moist fabric jacket, in which case a large part of the heat for evaporation comes from within.

In Fig. A of the reference mentioned, it is shown that the rate of drying of a moistened jacket on a metallic cylinder maintained at 98 F was substantially the same at 75.2 F 80 percent RH as it was at 82 F and 30 percent RH. Both conditions fall on the 73 ET line.

---

§A.S.H.V.E. RESEARCH REPORT No. 1321. Comfort Reactions of 275 Workers During Occupancy of Air Conditioned Offices, by Rowley, Jordan and Snyder. (See Chapter No. 1321.)

°Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, Cyril Tasker and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145.)

I cannot accept the statement that the effective temperature line stands solely on *instantaneous effects.* In another report[a] of a series of experiments, it is stated:

The same subjects took part in each of the eight 2-hr tests, four at 30 percent and four at 60 percent RH. To eliminate any effect of acclimatization to a temperature, when changing the conditions the upper and lower limits of the zone were determined by approaching them from opposite directions. Two general methods were followed in conducting the tests. First, the test was started with an atmospheric condition outside the comfort zone for most people, either below or above, and the probable center of the zone was approached by steps of 1 or 2 deg ET. Second, the test was started with a condition within the zone for most subjects, and then changed so as to move outward, to either a higher or a lower temperature.

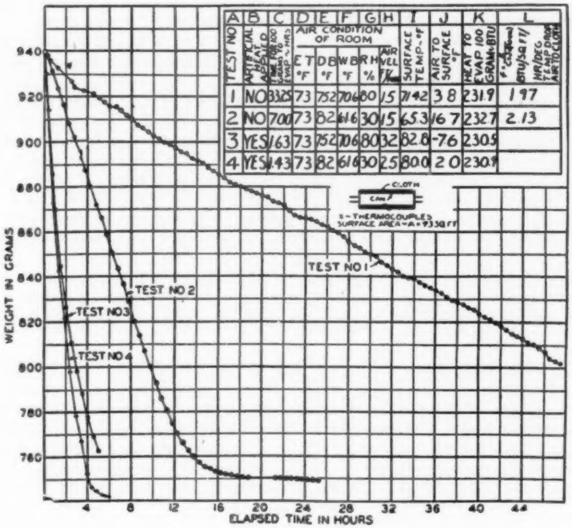


FIG. A. RESULT OF FOUR TESTS.

The range and direction of the change in temperature for the various tests are shown diagrammatically on the small psychrometric chart in Fig. 2.

This and subsequent papers by other investigators explored the possibilities of longer exposure and if, as Professor Yaglou says, the tolerance increases with time, then it is obvious that this is a factor which must be considered. It does not, however, per se prove that the original sensation is not an accurate index.

In all, though the possibility of errors due to sorption is not questioned, it would appear that adequate proof has not been advanced.

Based on References 5, 6 and 7, the author states that for winter or summer men and women were comfortable if their weighted skin temperature averages between 91.0 and 93.0 deg, regardless of the type of clothing or the environmental conditions.

[a] Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou. (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361.)

In all these papers the ultimate judgment of comfort is subjective. Reference 5 does not record humidity. Reference 6 maintained between 40 and 50 percent RH for all tests.

All of these papers present extremely valuable information but the real question is: Have we sufficient information available to justify substituting the indirect but objective measurement of skin temperature for a direct subjective test?

The smooth line for 50 percent RH in Fig. 2 indicates that a 1 deg change in ambient dry bulb would produce a 0.3 deg change in weighted average of skin temperature. Marked deviation from the smooth curve is noted for the points for 75 and 80 F dry bulb and, as between these two readings, change in skin temperature is 0.064 deg for each degree change in air temperature. For direct subjective observations, deviation of this magnitude might be expected, but these deviations cast considerable doubt on the validity of the objective method, which is intended to be an improvement on the subjective test. The paper does not state whether the points in Fig. 2 represent the average of the two individuals used in the tests. If they represent an average, then the deviation assumes even greater importance.

In the summary, the author makes three definite statements and one recommendation. The three statements do not appear to be substantiated by the presentation, as the proof is inadequate for either the failure of the present data or for the desirability of substituting the indirect objective skin temperature for the previous direct subjective method. As to the recommendation, there can be little question as to the advisability of further experimentation to correct or affirm the ET lines, particularly the determination of the winter optimum. Investigation of weighted skin temperature may be in order, provided the significance of the data is not assumed in advance of the experiments.

I suggest for consideration the possibility of a continuation of the type of experiment originally described by Ferry Houghten in the paper** previously mentioned, in the hope that it would provide a better theoretical background for the effective temperature concept.

ROBERT ARNOLD, Philadelphia, Pa. (WRITTEN): This paper presented by Professor Yaglou is, of course, well in line with his usual very careful study of this important subject. The paper is quite apropos, inasmuch as many things have come about since 1923 which should make it quite natural that the ET Index be investigated again.

It is quite important that Professor Yaglou has brought out the fact that exposed skin and clothing adsorb moisture and this adsorption, of course, imparts a sense of warmth when the subject passes from a relatively dry atmosphere to a moist and cooler one.

I am especially interested to note that Professor Yaglou has brought out the fact that no other index can compete with the ET Index in a certain definite region except when radiation effects are important. It seems to me that radiation effects have been very much overlooked in the past.

Undoubtedly the idea of mean skin temperature application represents a very rational method for correcting the present ET Index, and should be given further study.

R. W. KEETON, M. D., Chicago, Ill. (WRITTEN): After thinking over Professor Yaglou's paper I cannot see any serious objections to his conclusions.

In the first place, I think that he is correct in insisting that votes of comfort be taken after the subject is in the steady state. The entrance votes will certainly be influenced by the effect of heat input when moisture is added to the clothes, and, by heat loss when it is abstracted.

I believe that the second important point is the correlation of the skin temperature with comfort. The temperature of the skin is determined by the quantity of blood flowing through it. If heat is lost in evaporation or by radiation, the glomus shunts close down and reduce the flow. If the process is reversed, then they open and more blood flows through the skin. In a steady state, the blood flow will conceivably be such as to stimulate say x hot spots and y cold spots. It may shift so slowly that, only after the temperature of skin has been reduced, say 3 deg, will the number of cold

**Loc. cit. Note 9.

spots stimulated exceed greatly the number of hot spots. Then the subject votes *cold.* In a similar manner he will come to vote *hot or warm.*

It should, of course, be appreciated that the more rapidly the changes occur, the easier it will be to recognize the alterations in sensations. The production of cold sensation by the application of ether to the back of the hand is an illustration. A layer of clothes over the body slows the rate of heat exchange with the environment and makes it more difficult for one to arrive at a satisfactory vote expressing the gradation of comfort.

The correlation of the skin temperature with comfort should be made by a zone or a family of skin temperatures with experimentally determined limits. This would seem to be the answer to the question in the zone of vaso-motor regulation. For this reason, radiation effects and protection of clothing are most important modifying agents.

I should like to think that the regulation of the skin temperature is first a local one by way of local reflexes affecting the glomus bodies. This allows a minute effect to spread locally and gradually involve an area sufficiently large to produce a recognition of a sensory change.

ANDRÉ MISSENARD, Paris, France (WRITTEN): At the present stage of the effective temperature question two kinds of thermo-equivalent conditions must be considered: (1) thermal equivalences for sudden environmental changes, or instantaneous impressions, *(de passage),* and (2) thermal equivalences for a period of continuous exposure or adjustment *(de régime).*

The thermal equivalences of sudden change have been determined by Messrs. Houghten and Yaglou, and have led to the idea of the scale of effective temperatures.

When this research was resumed some fifteen years ago, we were induced to add another variable, that of the mean radiant temperature of the surrounding air, which by means of radiation exchanges combines its action with the temperature and velocity of the air. This brought us to the idea of *resultant temperature,* which is a generalization of the effective temperature. At that time (15 years ago), it was revealed that in a calm atmosphere, with practically motionless subjects, lowering the air temperature by one degree gave a corresponding rise in the temperature of the walls by one degree. Where there is a relative movement of the human body and of the air (whether due to the velocity of the air or due to the subjects' activity), the influence of radiation becomes perceptibly less and depends on this relative velocity of the body and of the air.

The definition of the equivalence of change *(de passage)* is simple and is based on the sensations of the subjects, or rather on the absence of sensation variations of the subjects passing, without transition, from one atmosphere to another.

Seeking to explain these results in terms of the laws of physics and of measurements made *under stationary conditions (en régime établi),* we found some discrepancies. Thus it became necessary to begin a systematic study of the laws of evaporation of the human body under stationary conditions. We found that these laws, which are laws of evaporation of inert bodies, are exactly the same as the laws of convection. In other words, *the coefficient of evaporation varies as a function of velocity exactly like the coefficient of convection.*

Knowledge of these laws permits the determination of the humidity coefficient of the skin, that is, the ratio between the quantity of moisture evaporated and that which would be evaporated if the body were wet over its entire surface.

When this humidity coefficient of the skin, based on the equivalence of change, is calculated, one finds a humidity coefficient of the skin quite superior to reality. This shows that in the equivalences of sudden change *(de passage),* the sensation of warmth depends not only on the surface of humidity of the skin, but also, as Mr. Yaglou emphasized in his paper, on the phenomena of absorption of water vapor by the skin and by surface hairs—phenomena which disappear under thermal conditions involved over a period of time necessary for adjustment.

We had, however, yet to find a correct definition for the equivalences involved in an adjustment period of time *(en régime établi)*. At the outset, like other authors, we believed it possible to define these equivalences by the uniformity of the mean temperature of the skin, this mean temperature being obtained by averaging the temperatures of the different parts of the body. Following the results of Dr. Winslow and his collaborators as well as the results which were repeated in our Saint-Quentin laboratory, we arrived at this conclusion: That as long as one deals in the zone of homeothermy (homéothermie), the mean temperature of the skin for unclothed subjects was, during a sustained period of time *(en régime)*, independent of the hygrometric degree. That is, the mean temperature of the skin depends only on the air temperature, on air velocity and on the mean radiant temperature of the walls.

Frankly, these measurements of skin temperature are always delicate and the results uncertain. But then, this conclusion is especially based on the fact that evaporation throughout a sustained adjustment period *(en régime)* is, within the limit of homeothermy, independent of the hygrometric degree. As measurements by weight are the most precise that can be made in this type of experiment, the conclusions drawn from it are the most valuable.

For motionless, unclothed bodies, the upper limit of homeothermy, in a calm atmosphere, is around 32 to 33 deg C for saturated air. We found that if the temperature is raised beyond this value and *if the hygrometric degree is lowered sufficiently to maintain homeothermy,* the temperature of the skin remains sensibly the same. Only evaporation increases to counterbalance the variation of loss or gain through convection or radiation; the degree of humidity of the skin adapts itself to the hygrometric degree of the atmosphere. When the homeothermal equilibrium is broken, that is, goes beyond 32 to 33 deg C for saturated air and about 50 deg for absolutely dry air, one can speak no longer of the temperature of the skin under sustained conditions, since it is no longer constant and since it (skin temperature) rises with time. We can denote the thermal equivalence of environments in established conditions *(en régime)* beyond the limit of homeothermy. We denote this equivalence of sustained or constant conditions variously: (*a*) by the temperature of the skin at the end of a determined time sufficient for adaptation; (*b*) by the elevation of the rectal temperature; or (*c*) by the rhythm of the heart at the end of a similar lapse of time.

Unfortunately, the sensation of comfort or discomfort of the subjects does not depend solely on the temperature of the skin even for a constant activity. It depends at the same time on its temperature and on its degree of humidity. Below 25 deg C, for unclothed bodies, the percentage of humidity of the skin is very small and is practically the sensation of warmth, and can be registered by the temperature of the skin. But as soon as the surrounding air temperature exceeds 25 deg, the degree of skin humidity becomes important and intervenes appreciably in the sensation of discomfort.

Near the upper limit of homeothermy *(homéothermie)*, when the skin temperature remains practically constant, it is this factor of skin humidity which most conditions the sensation of discomfort. It is difficult to analyze the reasons for this sensation of discomfort, just as it is difficult generally to analyze the ideas of fatigue. However, it is very necessary to recognize that it is nevertheless the factor which must be taken most into account; that is why at present, we resume our studies with the premise that the equivalences of sustained or steady state conditions *(en régime)* are based not only on the temperature of the skin but also on the subjective sensations of discomfort of different subjects.

The experiments to which we are presently devoted have shown that at an interval of one hour, the different subjects were able to compare their sensations. We are in the process of plotting these curves of equivalences of constant or sustained conditions *(en régime)* based on these sensations. It is a delicate task which will require much more time before becoming publishable. Yet it is these equivalences which

must be taken into account when comparing the various atmospheres in which men, and particularly the workers, must dwell.

The first results allow us to believe we shall thus arrive at an alignment chart with systems of curves similar to those of the effective temperatures made by Yaglou and Houghten. However (in our case) the humidity plays a practically negligible role below 25 deg; beyond this temperature, the humidity then rapidly increases, until at elevated temperatures we reach again, perceptibly, the scale of effective temperatures. This justifies the remark of Mr. Yaglou that, for equivalences of continuous exposure *(en régime)* the effective temperature regains all of its importance at elevated temperatures.

We are anxious to share the present direction of our studies with American scholars and engineers, hoping to receive whatever documents or suggestions which may be useful.

In closing, we wish to emphasize our pleasure in reading the paper by Professor Yaglou, with which we entirely agree, excepting perhaps the matters concerning the hygrometric degree's influence on the temperature of the skin within the limit of homeothermy.

Author's Closure: I wish to thank the gentlemen who discussed my paper, particularly the physicians and physiologists whose comments on body heat regulation should be of distinct value to the engineers.

Answering questions collectively, I would say that there are three ways in which humidity may affect skin temperature: (*a*) by its influence on evaporation from the skin, which becomes significant at high temperatures when the skin sweats, (*b*) by altering the thermal conductivity of clothing, and (*c*) through some local action on the respiratory mucosa capable of producing reflex changes of blood supply to the skin, particularly to the extremities. The relative importance of these factors varies greatly, and in all probability, the sum total effect is not a straight line function of humidity.

An important point which is not generally appreciated by engineers is that there are two forms of perspiration, *insensible* and *sensible*. The former is transudation of water through the blood capillaries and skin, and involves no sweat gland activity. Such insensible perspiration takes place at all temperatures below the comfort point, when there is no need for sweating. The amount of exudation is small and practically constant, irrespective of humidity, and, except in very low temperatures, regardless of temperature, also. Obviously, no more moisture can evaporate than is exuded, and variations of humidity should therefore have little or no effect, so long as the sweat glands remain inactive, as the late Mr. Houghten himself, among many others, had shown. Variations of humidity do affect evaporation from the lungs substantially in proportion to the vapor pressure difference, but the effect is small, since the amount of air breathed is only ¼ cfm in men at rest.

Sensible perspiration is the product of sweat glands, which begin to secrete in warm atmospheres when heat loss by radiation and convection is significantly reduced. The cooling of the human body by evaporation of sensible perspiration differs from that of an inanimate wet surface, in that the body itself controls the rate of sweating in accordance with requirements for heat balance. More sweat glands are thrown into action when the humidity is high than when it is low, in order to increase the wet surface and thus maintain needed evaporation in the face of decreasing vapor pressure differential. The limit of evaporative regulation is reached when the rate of sweating exceeds the evaporative capacity of the atmosphere. No one denies the vital importance of humidity in such hot atmospheres.

My paper deals primarily with possible humidity effects on insensible perspiration in the range of comfort heating and cooling, where the sweat glands are inactive, and where body heat regulation is achieved largely by changes of skin temperature. We need not worry about hot atmospheres at this time until we have first explored the ordinary air conditions in which we live and work. Although from the standpoint of body heat loss we are interested in the mean skin temperature of the whole body, from the standpoint of warmth or cold we are essentially concerned with changes of

mean skin temperatures in response to environmental changes. Attempts to correlate changes of forehead, cheek, hand or foot temperatures with sensations of warmth or comfort proved unsuccessful because different areas of the body responded differently to environmental changes.

Although the subjective feeling of warmth of an individual correlates well with his mean skin temperature, different individuals may differ as much as 3 F deg in skin temperature under identical conditions. This does not imply insensitivity of the skin. As Dr. Hardy has pointed out, a rise of but 0.0015 F deg of skin temperature in three seconds is sufficient to evoke a sensation of warmth, and a drop of 0.007 F deg in two seconds, a sensation of cold. Individual variations of skin temperature at least in part, seem to be related to differences in metabolic rates. As a rule, persons with low metabolic rates per unit of body surface area show lower mean skin temperatures than persons with high rates, as is to be expected, since their heat loss is also lower.

Reliance must therefore be placed on changes of mean skin temperature, in response to environmental changes, in determining air conditions that are alike in warmth. The subjects must be carefully chosen; they must be dressed alike, according to the season, and all skin temperature measurements must be made by one and the same method throughout.

The cost of the proposed study to the Society's Research Laboratory, covering conditions within the winter and summer comfort zones, and including radiation effects, should not exceed $12,000. With two observers, and five men and five women subjects, it will probably take about six months to complete the study, allowing from two to three months for the winter tests and an equal period for the summer tests.

1320

No. 1320

PHYSIOLOGICAL ADJUSTMENTS OF HUMAN BEINGS TO SUDDEN CHANGE IN ENVIRONMENT

By NATHANIEL GLICKMAN,* TOHRU INOUYE,** STANLEY E. TELSER, M.D.,*** ROBERT W. KEETON, M.D.,† FORD K. HICK, M.D.,†† CHICAGO, ILL., AND MAURICE K. FAHNESTOCK,††† URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Illinois, College of Medicine

THE physiological effect of different environmental temperatures and humidities on the human body has been studied extensively. The pattern of heat loss from the body under equilibrium conditions has been analyzed quantitatively over a rather wide range of environmental conditions [1,2,3,4]. On the other hand, little attention has been given to the adjustments of the human body to sudden changes in environmental temperature and humidity. The investigations of Houghten, et al [5,6] have dealt with both subjective and objective observations on normal subjects, clothed, entering an air conditioned space from the hot outside or after approximately 15 min in a hot room. The time required for the attainment of comfort or for the disappearance of perspiration was not measurably affected when the relative humidity was varied from 30 to 90 percent with the effective temperature remaining constant. Newton, Houghten, et al [7-9] have studied the subjective responses of a large group of workers after entering and leaving cooled and air conditioned offices as well as after one or more hours of occupancy. Sheard et al [10] have demonstrated the role played by the extremities of subjects passing from a comfortable environment to a cooler or warmer en-

*Physiologist, Aero-Medical and Atmospheric Environment Unit, Department of Medicine, University of Illinois. Member of A.S.H.V.E.

**Research Assistant, Department of Medicine, University of Illinois.

***Instructor in Medicine, Department of Medicine, University of Illinois.

†Head, Department of Medicine, University of Illinois. Member of A.S.H.V.E.

††Associate Professor of Medicine, Department of Medicine, University of Illinois.

†††Assistant Director, Engineering Experiment Station, Department of Mechanical Engineering, University of Illinois. Member of A.S.H.V.E.

[1]Exponent Numerals refer to References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Coronado, Calif., June 1947.

vironment. Hardy and Goodell [11] have just reported the results on subjects who were exposed for three hours to a warm environment (87.8 F) and then were conducted into an environment of 60.8-64.4 F for a four hour period. Metabolic rate, rectal and mean skin temperatures were measured and peripheral blood flow was computed. Upon exposure to cold, a temporary rise in rectal temperature was observed associated with a rapid decrease of the peripheral blood flow index.

General experience of the public has indicated that no ill effects are grossly apparent among normal individuals making the adjustments required by sudden cooling or sudden exposure to heat. This does not imply that these adjustments are accomplished with equal ease and without hazard by individuals with cardio-

Table 1. Description of the Experimental Subjects

Subject	Age Yrs	Height Ft & In.	Weight Lb	Area Sq Ft
B.F.	23	5- 5¾	152	19.0
J.B.	23	5-11¾	138	19.6
E.M.	22	6-2	150	20.8
W.P.	23	5-8	167	20.4
M.B.	19	5- 8½	130	18.4

vascular impairment. This problem and the effect of previous adaptation to heat or cold on these adjustments will be studied in the near future.

The experiments to be described were designed to study:

1. The rapidity and magnitude of the physiological adjustments made by the body on exposure to a sudden change in environmental temperature.

2. The effect of different relative humidities in altering the adjustments when passing from a hot to a comfortable environment.

3. The effect of different relative humidities on various physiological measurements with the body in approximate equilibrium with the environment.

Subjects and Experimental Conditions. Five healthy male medical students, ranging in age from 19 to 23 years (Table 1), were subjects for these experiments which were conducted from mid July to the end of September, 1946. Each subject participated in six experiments. They remained in the comfortable room for one hour with the relative humidity either at 30, 60 or 80 percent before and after the one or two hour hot room exposures.

Procedure. The subjects reported to the laboratory about one hour after the last meal, either at 8:00 a. m. or at 1:00 p. m., and always reported at the same time of day for each of the six experiments. They were instructed to drink three 8-ounce glasses of water before reporting, and were required to remain seated in the office for about 20 min before entering the comfortable room (designated Comfortable Room 1 or C.R. 1). Immediately on entering the comfortable room, the subject reported his *comfort* vote and began disrobing. He was weighed in the nude wearing only wooden sandals of known weight

and then dressed in the thermocouple union suit. He resumed his seat on the balance and remained sitting quietly until one hour had elapsed from the time of entrance into the comfortable room (Fig. 1). The subject then walked into the hot room and sat quietly in an identical balance for either one or two hours. After this he walked back into the comfortable room (designated Comfortable Room 2 or C.R. 2) and sat quietly in the first balance for one hour.

Experimental Rooms and Environmental Conditions. The comfortable room was 12 ft wide, 14 ft long and 8 ft high, completely air conditioned and well insulated. It was maintained at a constant dry bulb temperature of 76 F ± 0.5

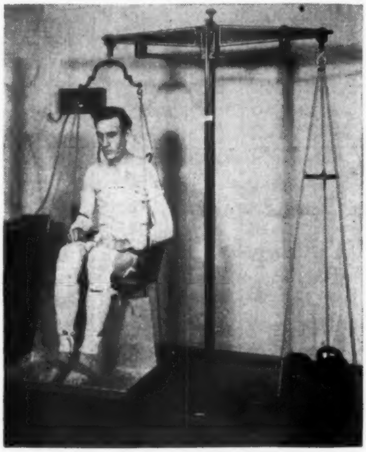


FIG. 1. SUBJECT WEARING THERMOCOUPLE UNION SUIT AND SEATED IN A TROEMNER BALANCE IN THE COMFORTABLE ROOM

deg with 30, 60 or 80 percent RH (68.8, 71.5 or 73.4 deg ET, respectively). The temperature of the environment, as measured by a globe thermometer, ranged between 76.2 and 76.6 F. Air velocity was constant and minimal and, as recorded by a vane type anemometer, was less than 25 fpm.

The hot room immediately adjoining the comfortable room was 19 ft wide, 21½ ft long, and 10 ft high. It was maintained at a dry bulb temperature of 98.5 F ± 1.0 deg with a 66 percent RH ± 4 percent (90.2 ET). The temperature of the environment, as measured by the globe thermometer, ranged between 97.2 and 98.8 F.

Equipment and Observations. Rectal and skin temperatures were obtained by a multiple lead constantan-copper thermocouple system leading to a potentiometer for the measurement of electromotive force [12]. The rectal thermocouple was inserted for a distance of 10 cm, as indicated by a rubber stop. Mean skin

TABLE 2. COMFORTABLE ROOM 1

Final Mean Skin and Rectal Temperatures, Average Rate of Evaporative Weight Loss, Comfort Vote, Pulse Rate and Blood Pressure

RH and ET[a]	30% and 68.8 ET				60% and 71.5 ET				80% and 73.4 ET				Comparison	Probability
	M[c]	σ[d]	R[e]	N[f]	M	σ	R	N	M	σ	R	N	Percent	
Mean Skin Temperature F (MT_s)	92.02	0.770	91.1 to 93.4	10	92.70	0.958	91.4 to 94.0	10	92.76	0.901	91.5 to 94.2	10	60 *vs* 30 80 *vs* 30 80 *vs* 60	0.005 0.006 Insign.
Rectal Temperature F (T_r)	97.88	0.379	97.2 to 98.3	10	98.22	0.498	97.4 to 99.0	10	98.45	0.450	97.8 to 99.2	10	60 *vs* 30 80 *vs* 30 80 *vs* 60	0.009 0.002 0.030
Evaporative Weight Loss[b] gm/min	0.92	0.296	0.62 to 1.59	10	1.06	0.404	0.52 to 1.80	10	0.96	0.337	0.57 to 1.74	10	60 *vs* 30 80 *vs* 30 80 *vs* 60	Insign. Insign. Insign.
Comfort Vote	4.2	0.33	4 to 5	10	4.4	0.45	4 to 5	10	4.3	0.46	4 to 5	10	60 *vs* 30 80 *vs* 30 80 *vs* 60	Insign. Insign. Insign.
Pulse Rate beats/min	72.0	10.77	60 to 92	10	79.1	11.01	60 to 93	10	79.0	10.44	60 to 96	10	60 *vs* 30 80 *vs* 30 80 *vs* 60	0.015 0.019 Insign.
Blood Pressure Systolic mm Hg	108.8	5.36	102 to 112	6	105.9	9.95	88 to 120	7	111.0		106 to 116	2	60 *vs* 30 80 *vs* 30 80 *vs* 60	Insign. Insign. Insign.
Diastolic mm Hg	71.7	6.10	60 to 80	6	72.3	6.11	64 to 82	7	78.0		72 to 84	2	60 *vs* 30	Insign.

[a]Relative Humidity and Effective Temperature
[b]Average rate during the last 40 min of an hour
[c]Mean values
[d]Standard deviation
[e]Range
[f]Number of observations

temperatures were computed by appropriately weighting values for 18 points on the skin [13]. A complete series of rectal and mean skin temperatures were obtained at intervals of 10 to 15 min, except during the adjustment period when they were recorded about every three minutes.

Two Troemner balances (Fig. 1) were used (sensitivity ± 0.5 gm at 80 kg load), one in each room, to obtain changes in weight of the subjects. The balances were modified by a wooden seat placed over the metal seat to reduce heat loss by conduction and a metal pan, replacing the foot rest, to catch any perspiration which might drip from the body. The weight of the subject was recorded at intervals of 3 to 4 min.

Pulse rate was obtained by palpation of the radial artery at the wrist. It was recorded at intervals of 10 to 15 min after the subject had been in either room for 20 min and at intervals of 3 to 4 min after the subject changed rooms. The blood pressure was obtained by auscultation using an aneroid sphygmomanometer and was recorded at the same intervals as the pulse rate.

The scale for the subjective sensation of comfort was similar to that used by Houghten, et al [5] and was as follows: 1—Cold; 2—Cool; 3—Slightly cool; 4—Comfortable; 5—Slightly warm; 6—Warm; 7—Hot.

The subjects were trained in the use of the comfort scale before starting the series of experiments. Intermediate votes such as 2½, 3½, or 4½ were accepted during the adjustment period.

Perspiration was judged to have started when tiny beads were first visible. Usually, these appeared earliest on the forehead, upper lip and hands.

Statistical Analysis of the Data. Student's [14,15] method and tables designated for determining the significance of the mean of a small series of paired differences were used.

RESULTS

Mean Skin Temperature

Comfortable Room 1. Freeman and Lengyel [16] have reviewed the conflicting results in the literature [10,17-21] on the effect of wide ranges in relative humidity on skin temperature. Some have reported results of observations on just a few points on the skin while others have reported on mean skin temperature. This report of Freeman and Lengyel details the results for many points on the body and reveals that in practically all instances the skin temperature is statistically significantly higher at the higher humidity. Mean skin temperatures have been calculated from the data given in Table 2 of their article. It was found to be 1.46 F and 1.11 F deg higher at 90 percent than at 20 percent RH when the dry bulb temperature of the environment was 75.2 and 89.6 F, respectively. The technical difficulties involved in maintaining a constant dry bulb temperature while raising the relative humidity may account for the greater increase in mean skin temperature at the lower environmental temperature.

Test data revealed that the relative humidity of Comfortable Room 1 had a distinct effect on the final mean skin temperature (Table 2). The final mean skin temperature of the subjects averaged 0.74 F deg higher at 80 than at 30 percent ($P = 0.006$) and averaged 0.68 F deg higher at 60 than at 30 percent ($P = 0.005$). No difference was observed between the values for 80 and 60 percent.

There was a slight increase in mean skin temperature averaging 0.43 F deg ($P < 0.0005$) from the time the subject put on his union suit and resumed his seat on the balance until the end of the period. The few minutes that the subject was in the nude permitted a decrease which was followed by the return to normal shortly after being clothed in the suit.

TABLE 3. HOT ROOM

Increase in Mean Skin and Rectal Temperatures (Deg F) from Final Values in Comfortable Room 1 and Time of Onset of Visible Perspiration

OBSERVATION	M[d]	s[e]	R[f]	N[g]
Δ MTs[a] at 10 Min	4.25	0.565	2.94 to 5.60	30
Δ MTs at 20 Min	4.34	0.612	2.96 to 5.78	30
Δ MTs at One Hour	4.35	0.726	2.75 to 5.84	30
Δ MTs at Two Hours	4.67	0.780	3.51 to 5.94	15
Max Δ Tr[b] within 10 Min	−0.18[c]	0.155	0.2 to −0.6	30
Δ Tr at One Hour	0.41	0.484	1.2 to −0.5	30
Δ Tr at Two Hours	1.05	0.492	2.0 to 0.2	15
Time of Onset of Visible Perspiration (Minute)	11.0	7.24	2.5 to 29.0	28

[a]Change in Mean Skin Temperature
[b]Change in Rectal Temperature
[c]Probability < 0.0005
[d]Mean values
[e]Standard deviation
[f]Range
[g]Number of observations

Hot Room. Reference to Table 3 and Figs. 2 and 3 shows that the major part of the redistribution of blood is accomplished in 2½ to 10 min after entering the hot room.

Comfortable Room 2. On entering Comfortable Room 2, following the one hour hot room exposure, the decrease in mean skin temperature (Table 4) at 10 min was significantly greater at 30 than at 80 percent RH ($P = 0.018$), and greater at 60 than at 80 percent ($P = 0.017$). There was no significant difference between the 30 and 60 percent values. This effect was still noticeable for the 20 min values.

Following the two hour hot room exposure the decrease in mean skin temperature in Comfortable Room 2 at 10 min, in all 3 comparisons, was significantly greater at the lower humidities. This effect was somewhat less consistent at 20 min.

It should be noted that the mean skin temperature decreased about the same amount at 10 and 20 min for comparable relative humidities in Comfortable

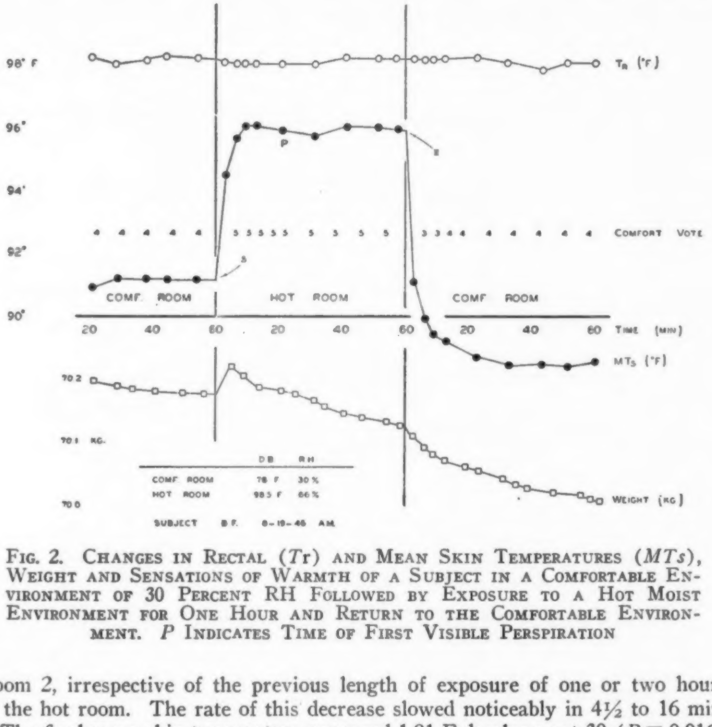


FIG. 2. CHANGES IN RECTAL (Tr) AND MEAN SKIN TEMPERATURES (MTs), WEIGHT AND SENSATIONS OF WARMTH OF A SUBJECT IN A COMFORTABLE ENVIRONMENT OF 30 PERCENT RH FOLLOWED BY EXPOSURE TO A HOT MOIST ENVIRONMENT FOR ONE HOUR AND RETURN TO THE COMFORTABLE ENVIRONMENT. P INDICATES TIME OF FIRST VISIBLE PERSPIRATION

Room 2, irrespective of the previous length of exposure of one or two hours in the hot room. The rate of this decrease slowed noticeably in 4½ to 16 min.

The final mean skin temperature averaged 1.91 F deg lower at 30 ($P = 0.016$) than at 80 percent RH and 1.49 F deg lower at 60 ($P = 0.004$) than at 80 percent. Thus, when moisture was present in the union suit (Comfortable Room 2) the effect of relative humidity in depressing the mean skin temperature was more evident than when the suit was relatively dry (Comfortable Room 1).

It may be concluded that the relative humidity of the environment is important in determining the rate and the extent of the decrease in mean skin temperature when perspiration is present on the skin and in the union suit.

Figs. 4 and 5 illustrate the changes of several different points on the surface of the body in making these adjustments. The time required for the skin temperature of a specific area to attain a comparative plateau after entering the hot room was prolonged when the entrance temperature was low. After all

points reached a plateau, the surface temperature of the body was relatively uniform. In Comfortable Room 2 a more rapid fall in skin temperature and lower final skin temperatures were observed in the environment having a relative humidity of 30 percent.

Rectal Temperature

Comfortable Room 1. The effect of different relative humidities on rectal temperature at comfortable dry bulb temperatures has received little attention.

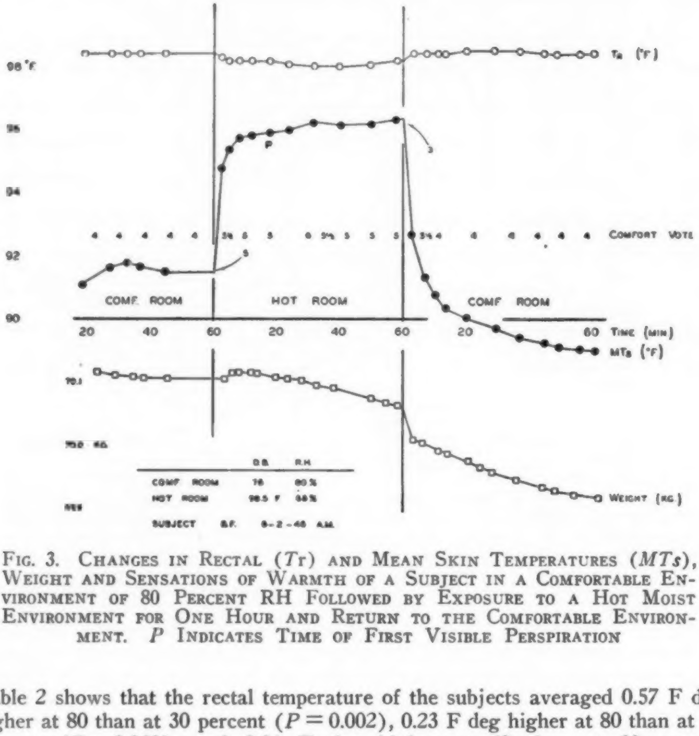


Fig. 3. Changes in Rectal (Tr) and Mean Skin Temperatures (MTs), Weight and Sensations of Warmth of a Subject in a Comfortable Environment of 80 Percent RH Followed by Exposure to a Hot Moist Environment for One Hour and Return to the Comfortable Environment. P Indicates Time of First Visible Perspiration

Table 2 shows that the rectal temperature of the subjects averaged 0.57 F deg higher at 80 than at 30 percent ($P = 0.002$), 0.23 F deg higher at 80 than at 60 percent ($P = 0.030$), and 0.34 F deg higher at 60 than at 30 percent ($P = 0.009$). There seems little doubt that the relative humidity had a distinct effect on the final rectal temperature.

In all but one experiment the rectal temperature fell during the one hour in Comfortable Room 1 ($P < 0.005$). The average decrease was 0.38 F deg and the maximum was 1.0 F deg. The increase which occurred in the one case amounted to only 0.1 F deg. This decrease in rectal temperature has been noted by others [4] and may be ascribed to the reduction in heat production which occurs when the subject relaxes on the balance.

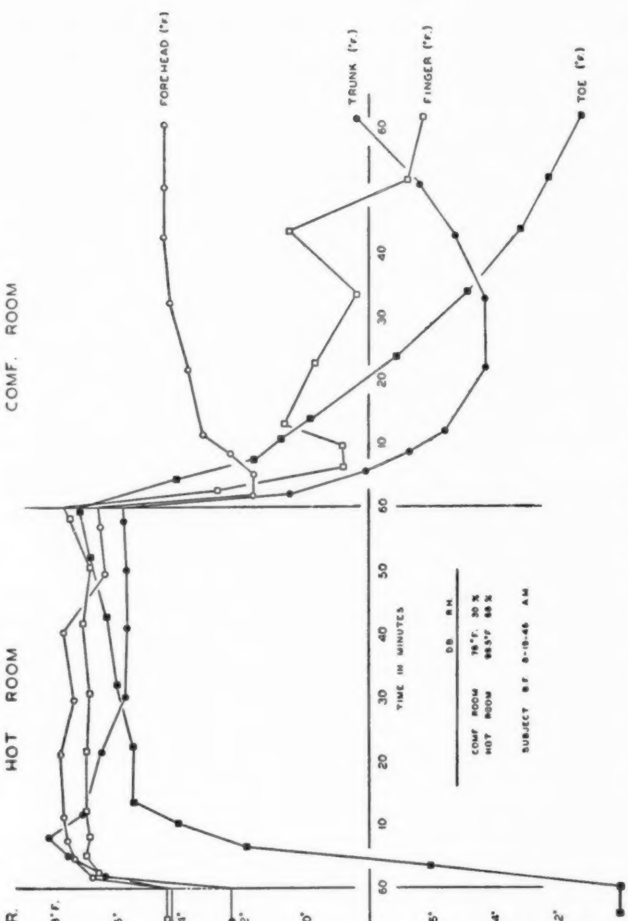


FIG. 4. CHANGES IN SKIN TEMPERATURES OF FOUR SITES ON THE BODY OF A SUBJECT (TRUNK REPRESENTS MEAN OF 6 POINTS) IN A COMFORTABLE ENVIRONMENT OF 30 PERCENT RH FOLLOWED BY EXPOSURE TO A HOT MOIST ENVIRONMENT FOR ONE HOUR AND RETURN TO THE COMFORTABLE ENVIRONMENT

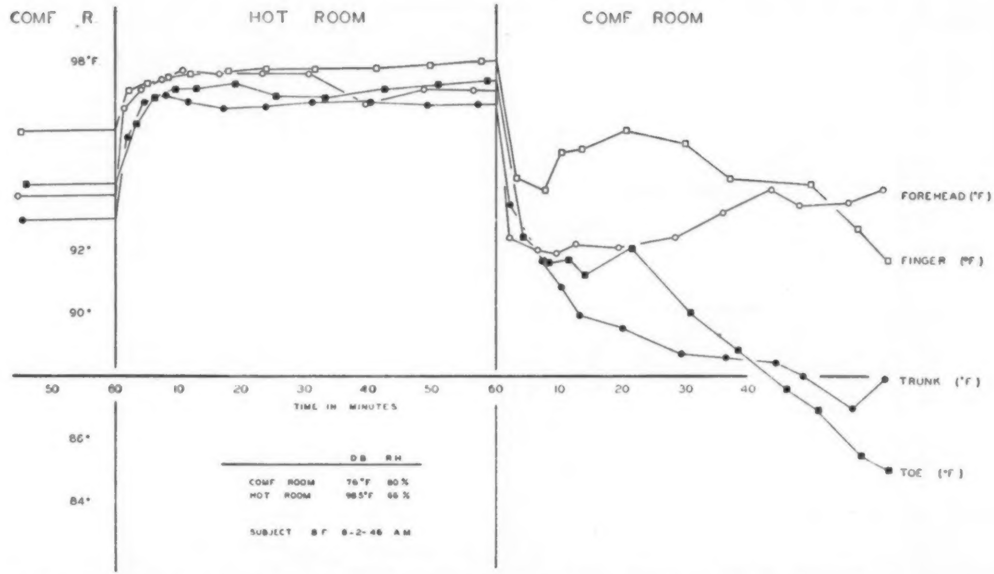


FIG. 5. CHANGES IN SKIN TEMPERATURES OF FOUR SITES ON THE BODY OF A SUBJECT (TRUNK REPRESENTS MEAN OF 6 POINTS) IN A COMFORTABLE ENVIRONMENT OF 80 PERCENT RH FOLLOWED BY EXPOSURE TO A HOT MOIST ENVIRONMENT FOR ONE HOUR AND RETURN TO THE COMFORTABLE ENVIRONMENT

Hot Room. It has been noted how rapidly the redistribution of blood occurs, as revealed by changes in mean skin temperature when a subject enters a hot room after having been in a comfortable environment and vice versa. In preliminary studies, discontinued in 1941, a statistically significant increase in rectal temperature was noted when subjects entered a comfortable environment after having been in a hot environment for one or two hours. The reverse was observed in a few experiments performed when the subjects entered the hot room after having been in a comfortable environment for one hour. This observation has been reported by others [11,22-26].

Test results (Table 3) showed that within 10 min of entering the hot room a small but true decrease in rectal temperature occurred. The extent of this decrease was not affected by the relative humidity of Comfortable Room 1. The maximum decrease recorded was 0.6 F deg, and the average was 0.18 F deg ($P < 0.0005$). On entering the hot room there is a transfer of heat from the deep to the superficial tissues as a result of the rapid redistribution of blood.

It required an average of 19.1 min for the rectal temperature to begin increasing and an average of 25.6 min to return to the level observed immediately prior to entrance. Thus, during the first 20-25 min, part of the heat, instead of being dissipated into the environment, was being utilized to warm the superficial tissues. During this period the body was completing the adjustments required to establish equilibrium with the environment, including increasing activity of the sweat glands.

After the initial decrease was completed the average rate of increase was 0.92 F deg per hr for the remainder of the hour and 0.55 F deg per hr for the second hour. This difference was significant ($P < 0.0005$) and indicated that the heat loss from the body was more rapid during the second hour.

After one hour in the hot room the rectal temperature averaged 0.41 F deg higher than the final Comfortable Room 1 value and after two hours in the hot room averaged 1.05 F deg higher.

Comfortable Room 2. A small but definite increase in rectal temperature was observed within 10 min of entering Comfortable Room 2, confirming the authors' unpublished results of 1941 (Table 4). As has been mentioned, similar results have been obtained by Shoji and Ogata [26] and Hardy and Goodell [11]. This increase occurred whether the exposure had been for one or two hours in the hot room and averaged 0.19 F deg ($P < 0.001$) and 0.15 F deg ($P = 0.0006$) for the one and two hour hot room exposures, respectively. The maximum increase recorded was 0.5 F deg. Further, it should be noted that this increase occurred to the same extent whether the relative humidity in Comfortable Room 2 was high or low.

The rectal temperature decreased 0.29 F deg more ($P = 0.006$) after a two hour than after a one hour hot room exposure, although the final rectal temperature averaged 0.30 F deg higher ($P = 0.0015$) after the two hour exposure. Since, in most instances, the union suits contained more moisture after two hours in the hot room, the subjects had a greater heat loss after the longer exposure to heat.

Time of Observation of First Visible Perspiration in Hot Room. The earliest that perspiration was observed was 2½ min, the latest 20 min after entering the

TABLE 4. COMFORTABLE ROOM 2

Decrease in Mean Skin and Rectal Temperatures (Deg F) From Final Values in Hot Room After One Hour in Hot Room

OBSERVATION	M^a	σ^b	R^c	N^d	COMPARISON PERCENT	PROBABILITY
△ MTs at 10 min in 30% RH....	6.60	0.804	5.20 to 7.45	15	30 *vs* 80	0.018
△ MTs at 10 min in 60% RH....	6.33	0.550	5.59 to 7.09	15	30 *vs* 60	Insign.
△ MTs at 10 min in 80% RH....	5.29	0.452	4.56 to 5.96	15	60 *vs* 80	0.017
△ MTs at 20 min in 30% RH....	7.30	1.139	5.21 to 8.50	15	30 *vs* 80	0.009
△ MTs at 20 min in 60% RH....	7.36	0.495	6.66 to 7.95	15	30 *vs* 60	Insign.
△ MTs at 20 min in 80% RH....	6.19	0.581	5.13 to 6.78	15	60 *vs* 80	0.027
Max △ Tr within 10 min..........	−0.19	0.115	0.4 to 0.0	15	Final Hot Room Tr *vs* Max Tr <0.001 within 10 min in Comfortable Room 2	
AFTER TWO HOURS IN HOT ROOM						
△ MTs at 10 min in 30% RH....	6.92	0.574	6.02 to 7.59	15	30 *vs* 80	0.001
△ MTs at 10 min in 60% RH....	6.06	0.281	5.51 to 6.29	15	30 *vs* 60	0.001
△ MTs at 10 min in 80% RH....	5.18	0.424	4.59 to 5.65	15	60 *vs* 80	0.018
△ MTs at 20 min in 30% RH....	8.35	0.789	7.14 to 9.61	15	30 *vs* 80	<0.001
△ MTs at 20 min in 60% RH....	7.25	0.359	6.70 to 7.70	15	30 *vs* 60	0.037
△ MTs at 20 min in 80% RH....	6.25	0.544	5.39 to 6.79	15	60 *vs* 80	0.045
Max △ Tr within 10 min..........	−0.15	0.135	0.5 to −0.1	14	Final Hot Room Tr *vs* Max Tr 0.001 within 10 min in Comfortable Room 2	

[a]Mean value
[b]Standard deviation
[c]Range
[d]Number of observations

hot room. One subject perspired early in all six of his experiments (2½ to 4½ min) while on another subject perspiration appeared as early as 10 min and as late as 29 min. Thus, the time of appearance of visible perspiration varied from one subject to another and even in the same subject from one experiment to another.

The relative humidity in Comfortable Room 1 had a definite effect on the time of onset of visible perspiration in the hot room. When the subjects had previously been exposed to an 80 percent RH in Comfortable Room 1, the average time for the onset of visible perspiration was 6.8 min less ($P = 0.004$) than when the previous exposure had been 30 percent. It is interesting to note that both the final rectal and mean skin temperatures were higher at 80 percent than at 30 percent RH in Comfortable Room 1. In short, the subjects were nearer the zone of evaporative regulation when in the environment having an RH of 80 percent.

Comfort Vote

Comfortable Room 1. The subjects and observers noted on entrance that Comfortable Room 1 at 80 percent RH felt slightly less comfortable (on the warm side of the scale) than at the other humidities. This difference was too slight to alter the comfort vote when no perspiration was present on the skin. The final comfort vote in Comfortable Room 1 was unaffected by the relative humidity.

In 23 of the 30 experiments the subjects were comfortable at all humidities. Two of the five subjects felt slightly warm (voted 5) in 7 experiments (3 at 80, 3 at 60, and 1 at 30 percent) and, of the two, one felt slightly warm in 5 of his 6 tests. Thus, all of the slightly warm votes, after one hour in Comfortable Room 1, were given by two subjects. These results, therefore, represent differences of individuals rather than differences due to the environment.

Hot Room. The sensation of warmth obtained on entering the hot room showed individual variations and bore no relationship to the relative humidity of Comfortable Room 1 from which the subjects had entered. Two of the five subjects voted 5 (slightly warm) in all but one of their experiments; one subject voted 7 (hot) in all but one experiment and one subject voted 7 in 2 experiments. So, on entrance into the hot room, there were 11 votes of 5, 12 votes of 6 and 7 votes of 7.

The final hot room votes also represent differences of individuals. The two subjects who consistently voted 5 on entrance gave final votes of 5 or 6 in all but 2 experiments. The other three subjects voted 7 in all but one experiment.

Comfortable Room 2. The votes on entrance into Comfortable Room 2 varied from 1 to 3, with one exception. One subject (W.P.) who always perspired freely in the hot room voted 0. In 24 experiments the comfort vote during the first minute remained the same as the entrance vote. However, two subjects in 6 experiments lowered their vote indicating that they felt still cooler.

There are a number of factors which would presumably affect the degree of coolness experienced by healthy persons on entering a comfortable environment of constant dry bulb temperature from a hot environment. These include the

amount of moisture present on the skin and in the clothing, the feeling of warmth, the level of skin and rectal temperature and the relative humidity of the comfortable environment.

Houghten et al [8] reported *a reasonably good correlation* between the degree of perspiration experienced before entering a cooled space and the comfort vote on entrance (correlation coefficient of –0.47 as calculated by the authors from data shown in Fig. 10 of their paper.) Test subjects were fewer in number and exposed to a slightly higher effective temperature for a longer period of time in the hot room. Nevertheless, no correlation could be demonstrated between the comfort vote and the amount of moisture present on the skin and in the union suit at the time of entrance into Comfortable Room 2 (see Table 7). It is difficult to visualize why such a relationship per se should exist, since the comfort vote is given immediately on entrance and this is too soon for evaporative heat loss to become the dominant factor.

Houghten et al[8] also reported *a reasonably good correlation* between the degree of warmth before entering a cooled space and the comfort vote on entrance. The correlation coefficient for his data, as calculated by the authors is –0.53 and is –0.50 for their data. This suggests that the higher the vote in the hot room the lower will be the vote on entrance to a comfortable environment.

From the authors' test data it was expected that no relationship would exist between the final mean skin temperature in the hot room and the sensation of coolness on entering Comfortable Room 2, since all values were within a narrow range of 1.5 F deg. It is reasonable to expect that with a greater range in mean skin temperature such a relationship might become apparent. There was also no relationship between the sensation of coolness on entering Comfortable Room 2 and either the final rectal temperature in the hot room or the relative humidity in Comfortable Room 2. There are undoubtedly many factors inherent in the individual as well as possible combinations of external factors which influence the vote and are difficult to interpret. At this time a more extensive analysis of the data has not been undertaken.

The data on the time required for the attainment of comfort in Comfortable Room 2, after the hot room exposures, were difficult to analyze because in many of the experiments a vote of 4 was never attained. Therefore, the data were examined in a manner which would yield information as to the relative effect of the three conditions studied on the time the votes of 3, 3½, and 4 were obtained. If, for example, a subject voted 3 after 16 min in Comfortable Room 2 at 80 percent and required 22 min at 30 percent RH, it was considered that the environment of 80 percent RH had a favorable effect. The data for the 1 and 2 hr hot room experiments were combined and this type of comparison was made for the relative effect of the three humidity levels.

Of the 28 comparisons for the 3 environmental conditions in which a vote of 3 was obtained, 21 favored the higher relative humidities and 7 favored the lower relative humidities, a deviation of 7 from the ideal outcome of a random pattern. The standard deviation of the distribution of 28 events that may occur with equal probability in either of two ways is $\sqrt{0.5 \times 0.5 \times 28} = 2.646$. The probability is 0.014 that such an outcome as was observed would have resulted from the operation of random factors only.

Of the 28 comparisons for the 3 environmental conditions in which a vote of 3½ was obtained, 23 favored the higher relative humidities and 5 favored the lower relative humidities, a deviation of 9 from the ideal outcome of a random pattern. For the same number of comparisons obtained for the vote of 4 (when a subject did not vote 4 his response at the next lower vote was used), 21 favored the higher and 7 favored the lower relative humidities, a deviation of 7 from the ideal outcome of a random pattern. The probability that such an outcome as was observed would have resulted from the operation of random factors is 0.004 and 0.014, respectively. Thus, the data yield strong evidence that relative humidity does play a role in the rapidity of attainment of comfort under the experimental conditions imposed.

Mean Skin Temperature at the Time of Comfort Vote of 4. The relation of the sensation of warmth to skin temperature has been studied by a number of observers. Miura [18], Phelps and Vold [20], Ward [27] and Houghten et al [6] have made measurements on a few selected sites on the skin while Winslow et al [28,29], Hardy et al [30], Yaglou et al [31], and this laboratory [32] have made observations on mean skin temperature. Ward [27] has noted the difficulty in comparing the results of different investigators because of differences in sites measured, as well as in methods employed. Nevertheless, there is a uniformity of opinion that warmth sensations are definitely related to skin temperatures. It is believed that mean skin temperature is a better index than any specific point since specific points, particularly the extremities, show far greater variations from day to day and from subject to subject under the same environmental conditions. For example, low temperatures of the hands and feet have been recorded while subjects were comfortable (Fig. 4).

Winslow et al [28] reported the comfort votes to be on the pleasant side when the mean skin temperature ranged from 88.0 to 92.9 F. Hardy et al [30] reported mean skin temperatures between 91.4 and 93.5 F in the zone of vasomotor regulation, and Yaglou et al [31] reported that *comfort was associated with mean skin temperatures between 91.5 and 93.5 F in both men and women regardless of air temperature which varied from 53.5 to 84 F, amount of clothing worn (0 to 14.5 lb) and season of the year.* This laboratory has reported [32] that the mean skin temperature ranged from 92.3 to 93.8 F when the subjects were nude and under basal conditions in an environment in which they felt comfortable.

In the present study we have found a range in mean skin temperature of 91.1 to 94.1 F to be associated with comfort votes of 4 in Comfortable Room 1. Votes of 4½ or 5 with values below 94 F have also been observed.

In Comfortable Room 2, after the hot room exposure, the mean skin temperature ranged from 87.6 to 92.4 F when the subjects were voting 4. This range was practically identical, 87.6 to 92.3 F, at the time the subjects were voting 3½. Further, each subject showed a somewhat greater range in mean skin temperature in Comfortable Room 2 for a vote of 4 than in Comfortable Room 1.

It should be noted that the mean skin temperature alone cannot be used as an index of the subjective sensation of warmth during the early adjustment period when the subject passes from a hot environment to a comfortable environment or vice versa. This is due to the sudden and rapid change in the rate of heat exchange that occurs during this period.

TABLE 5. PULSE RATE AND BLOOD PRESSURE ADJUSTMENTS

	Hot Room						Comfortable Room 2					
	Comparison	D^a	σ	R	N^b	P^g	Comparison	D^a	σ	R	N^b	P^g
Pulse Rate beats/min	First P.R.[c] *vs* Final C.R. 1[d] P.R.	7.7	6.12	24 to −4	30	<0.0005	First P.R. *vs* Final Hot Room P.R.	−11.6	7.37	3 to −27	30	<0.0005
	One Hour P.R. *vs* Final C.R. 1 P.R.	10.2	7.54	26 to −4	30	<0.0005	Second P.R. *vs* First P.R.	1.6	4.18	12 to −6	30	0.025
	One Hour P.R. *vs* First P.R.	2.5	6.96	16 to −10	30	0.033	First P.R. at 80% *vs* First P.R. at 30%	−6.5	8.12	6 to −21	10	0.021
	Two Hour P.R. *vs* First P.R.	8.9	5.96	20 to −10	14	<0.0005	First P.R. at 60% *vs* First P.R. at 30%	−4.1	5.67	7 to −15	10	0.030
	Two Hour P.R. *vs* One Hour P.R.	5.1	4.39	14 to −2	14	0.0007	First P.R. at 80% *vs* First P.R. at 60%	−3.1	10.39	17 to −19	10	Insign.
							Final P.R. *vs* P.R. in C.R. 1 (1 Hour Hot Room Exposure)	−4.6	4.41	4 to −11	15	0.001
							Final P.R. *vs* Final P.R. in C.R. 1 (Two Hour Hot Room Exposure)	−2.8	6.09	8 to −17	15	0.055

TABLE 5 *(Continued)*. PULSE RATE AND BLOOD PRESSURE ADJUSTMENTS

	HOT ROOM					
	COMPARISON	D^a	σ	R	N^b	P^g
Systolic Pressure mm Hg	First S.P.[e] *vs* Final C.R. 1 S.P.	6.4	7.13	21 to −12	14	0.004
	Second S.P. *vs* First S.P.	−9.9	14.27	20 to −36	14	0.013
	Lowest S.P. *vs* Final C.R. 1 S.P.	−14.1	10.93	2 to −32	14	<0.0005
	One Hour S.P. *vs* Final C.R. 1 S.P.	−0.2	6.67	12 to −11	12	Insign.
Diastolic Pressure mm Hg	First D.P.[f] *vs* Final C.R. 1 D.P.	−3.1	4.71	6 to −14	14	0.019
	Second D.P. *vs* Final C.R. 1 D.P.	−6.1	8.57	2 to −31	14	0.013
	Lowest D.P. *vs* Final C.R. 1 D.P.	−11.9	8.01	0 to −31	14	<0.0005
	One Hour D.P. *vs* Final C.R. 1 D.P.	−3.0	4.30	2 to −14	12	0.022

[a]Mean of the differences obtained for the designated comparison. Plus indicates that the first listed observation had a higher value.
[b]Number of cases
[c]Pulse rate
[d]Comfortable room 1
[e]Systolic pressure
[f]Diastolic pressure
[g]Probability.

Pulse Rate

Comfortable Room 1. Table 2 shows that the final pulse rate averaged 72.0, 79.1 and 79.0 for the environmental conditions of 30, 60 and 80 percent RH, respectively. The difference between the rates for the comparison of 30 percent with 80 percent was definite ($P = 0.019$) as was the difference for the comparison of 30 percent with 60 percent ($P = 0.015$). This is not to be in-

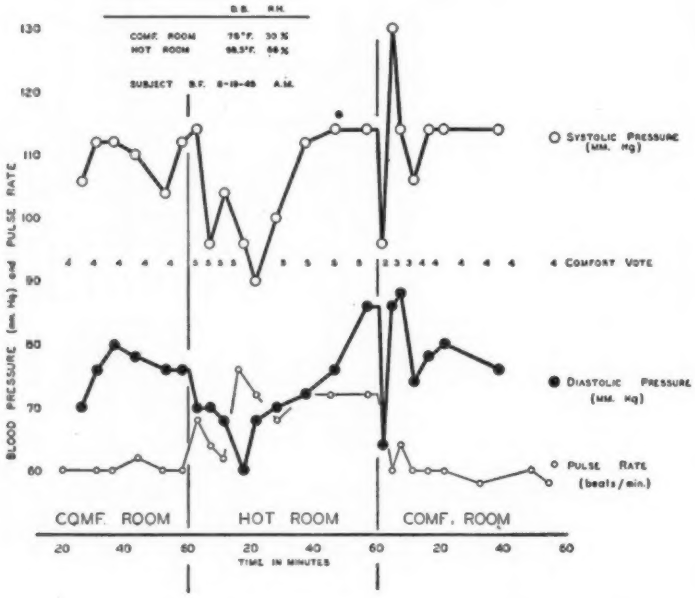


FIG. 6. CHANGES IN BLOOD PRESSURE, PULSE RATE AND SENSATION OF WARMTH OF A SUBJECT IN A COMFORTABLE ENVIRONMENT OF 30 PERCENT RH FOLLOWED BY EXPOSURE TO A HOT MOIST ENVIRONMENT FOR ONE HOUR AND RETURN TO THE COMFORTABLE ENVIRONMENT. THIS GRAPH ILLUSTRATES THE LARGE FLUCTUATIONS IN BLOOD PRESSURE OCCASIONALLY OBSERVED

terpreted as a direct effect of relative humidity on the pulse rate but rather as a result of the increase in body temperature. Keeton et al [33] have shown a high degree of correlation between pulse rate and rectal temperature under more or less equilibrium conditions. Reference to Table 2 reveals that for these same two comparisons referred to previously, both the final rectal and mean skin temperatures were significantly higher at the higher humidities.

Table 2 also shows the range in pulse rate obtained for the different subjects. It should be noted that the lowest values, 60, were given by one subject and the highest values, 92 to 96, by another subject.

Hot Room. The literature contains many reports on the effect of varying environmental temperatures and humidities on pulse rate[34-38]. Haldane[34] reported an increase of 20 beats of the heart per minute for every 1 F deg rise in rectal temperature for subjects exposed in a hot room or in a Turkish bath.

An increase in pulse rate was observed on entering the hot room (Table 5, Figs. 6, 7). This increase was not related to the final rate in Comfortable Room 1, since the increase was about the same for the subjects with low pulse rates as it was for the subjects with high pulse rates. The increase averaged 7.7

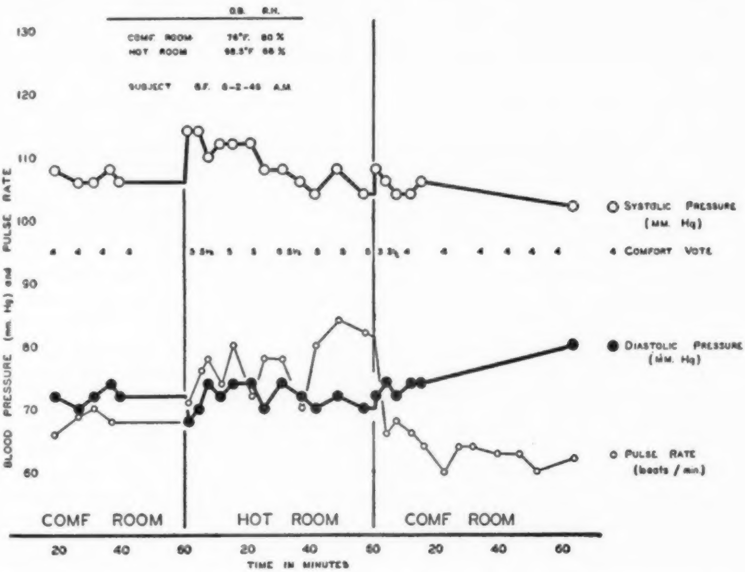


FIG. 7. CHANGES IN BLOOD PRESSURE, PULSE RATE AND SENSATION OF WARMTH OF A SUBJECT IN A COMFORTABLE ENVIRONMENT OF 80 PERCENT RH FOLLOWED BY EXPOSURE TO A HOT MOIST ENVIRONMENT

beats per minute ($P < 0.0005$) with a range of 24 to –4. There was no significant difference between the first and second pulse rate values taken after entrance into the hot room. The average time for obtaining these values was 4.1 and 8.1 min, respectively.

At the end of one hour in the hot room the pulse rate averaged 10.2 beats per minute faster ($P < 0.0005$) than the final Comfortable Room 1 value with a range of 26 to –4. It should be noted that the one hour pulse rate averaged only 2.5 beats per minute faster than the first value obtained in the hot room ($P = 0.033$).

In order to be certain that this increase in pulse rate, which occurred on entering the hot room, was not due to the muscular effort involved in walking into

TABLE 6. AVERAGE RATE OF EVAPORATIVE WEIGHT LOSS IN COMFORTABLE ROOM 1
(grams per minute)

SUBJECT	RELATIVE HUMIDITY PERCENT						MEAN RATE	MEAN RATE PER M^2
	30	60	80	30	60	80		
B.F.	0.62	0.60	0.60	0.65	0.78	0.57	0.64	0.36
J.B.	0.88	0.97	1.17	0.70	1.17	1.06	0.99	0.55
E.M.	0.79	0.52	0.80	1.05	0.70	1.00	0.81	0.42
W.P.	1.19	1.80	1.74	1.59	1.57	1.18	1.51	0.80
M.B.	0.62	1.11	0.70	1.05	1.36	0.77	0.93	0.55

TABLE 7. APPROXIMATION OF MOISTURE (GRAMS) PRESENT ON SURFACE OF SKIN AND IN UNION SUIT AT TIME OF ENTRANCE INTO COMFORTABLE ROOM 2[a]

SUBJECT	ONE HOUR HOT ROOM EXPOSURE			TWO HOUR HOT ROOM EXPOSURE		
	RH OF COMFORTABLE ROOM PERCENT					
	30	60	80	30	60	80
B.F.	133	264	184	350	400	218
J.B.	105	258	313	358	276	477
E.M.	176	262	187	397	214	311
W.P.	271	345	318	479	405	683
M.B.	224	284	140	232	355	374

[a]These values were computed as follows: (1) Obtain amount lost in 60 min in comfortable Room 1; (2) Substract (1) from total evaporative weight loss for 60 min in comfortable Room 2; (3) Add residual moisture in union suit at termination of the experiment.
This assumes (possibly incorrectly) that the insensible weight loss is not altered by a wet covering on the skin. The error introduced by this assumption, regardless of other factors involved, will be small.

TABLE 8. HEAT LOSS BY EVAPORATION (CAL/M²/HR) DURING THE FIRST AND SECOND HOURS OF THE TWO HOUR HOT ROOM EXPOSURE

SUBJECT	FIRST HOUR			SECOND HOUR		
	RH IN COMFORTABLE ROOM PERCENT					
	30	60	80	30	60	80
B.F.	19.5	21.4	29.8	43.1	35.7	30.8
J.B.	30.6	31.7	38.1	53.9	62.0	46.9
E.M.	7.7	25.9	24.8	35.8	35.8	36.4
W.P.	28.4	44.3	43.6	40.3	36.3	56.5
M.B.	37.2	42.3	28.7	50.5	56.6	51.6

the adjoining room and getting seated on the balance, the experiments were repeated but the two rooms were kept at identical environmental conditions (76 F dry bulb, 30 percent RH). No demonstrable change in pulse rate occurred under these conditions when the subjects changed rooms and remained seated on the balance for 15 min. Thus, the initial increase in pulse rate was primarily due to the vascular changes involved in the redistribution of blood which occurred within the first few minutes in the hot room. Further, as will be discussed later, a decrease in pulse rate occurred within a few minutes of entering Comfortable Room 2.

The pulse rate showed a further increase of 5.1 beats per minute ($P < 0.001$) between the first and second hour values in the hot room. This increase may justifiably be attributed to the increase in body temperature which occurred during this period.

Comfortable Room 2. Haldane [34] and Adolph and Fulton [38] have reported an immediate decrease in pulse rate when subjects entered a cool room after having been exposed to hot conditions. The authors' data confirm these observations.

The average decrease in pulse rate on entering Comfortable Room 2 for all 30 tests was 11.6 beats per minute ($P < 0.0005$) with a range of 27 to –3. The extent of this decrease was the same whether the subjects had been in the hot room for one or two hours. However, the relative humidity of Comfortable Room 2 definitely affected the extent of this decrease. The pulse rate decreased 6.5 beats per minute more at 30 than at 80 percent relative humidity ($P = 0.021$), 4.1 beats per minute more at 30 than at 60 percent ($P = 0.030$) and 3.1 beats per minute more at 60 than at 80 percent RH, although this difference was not statistically significant.

After an exposure of one hour in the hot room, the final pulse rate obtained in Comfortable Room 2 averaged 4.6 beats per minute less ($P = 0.001$) than the final pulse rate in Comfortable Room 1 (Figs. 6, 7). A similar comparison after a two hour hot room exposure revealed no significant differences. The final rectal temperature in Comfortable Room 2, as has been mentioned, was higher following the two hour hot room exposure.

Blood Pressure

Comfortable Room 1. In most respects the changes in blood pressure, if any, were slight. The relative humidity had no effect on either the systolic or diastolic pressure.

Hot Room. Within two minutes of entering the hot room, there was a definite increase ($P = 0.004$) in systolic pressure of 6.4 mm Hg (range of 21 to –12) and a true decrease ($P = 0.019$) in diastolic pressure of 3.1 mm Hg (range 14 to –6). On the second reading, taken about 5.5 min after entrance, the systolic pressure declined 9.9 mm Hg ($P = 0.013$) from the initial reading while the diastolic pressure remained unchanged. Fig. 7 illustrates the occasional large fluctuations in blood pressure which may occur.

The decrease in diastolic pressure reflects a slight reduction in peripheral resistance brought about by the marked vasodilatation which occurs on entering the hot room. The transient initial increase in systolic pressure is unexplained. It may be due to psychic factors associated with anticipation of an unpleasant experience on entering a hot moist environment.

The greatest decrease in systolic and diastolic pressure occurred in about 12 to 15 min (Figs. 6, 7) and averaged 14.1 mm Hg (range of 32 to –2, $P < 0.001$) and 11.9 mm Hg (range of 31 to 0, $P < 0.001$), respectively. The time for this to occur was rather consistent for each subject. After one hour in the hot room, the systolic pressure had returned to the same level observed in Comfortable Room 1 while the diastolic pressure was only 3.1 mm Hg lower ($P = 0.022$). Information on the change in blood pressure between the first and second hours of a two hour hot room exposure is inconclusive because of too few observations. Further experiments are planned to elucidate this point.

Comfortable Room 2. No significant changes occurred in either systolic or diastolic pressure on entering, during, or after one hour.

Evaporative Weight Loss

Comfortable Room 1. It is agreed that the weight loss from total insensible perspiration is brought about primarily by the evaporation of water from the respiratory passages and skin and to a slight extent by the gaseous exchange of carbon dioxide and oxygen. Factors which may affect the insensible perspiration are the temperature, vapor pressure and air motion of the environment, the blood flow [39] and temperature of the skin [25,39] and the metabolic rate [40]. The data reported here were evaluated to determine whether the vapor pressure of the ambient air significantly altered total insensible perspiration.

The insensible weight loss was observed during approximately the last forty minutes of the one hour Comfortable Room 1 exposure period. Weight loss for the individual experiments (Figs. 2, 3) was quite constant during this period and the results are presented in Table 6.

For some subjects the variation from one experiment to another may be large. Subject M.B. (Table 6) had average rates ranging from 0.62 gram per min to 1.36 gram per min. On the other hand, subject B.F. had average rates ranging only from 0.57 to 0.78 gram per min. Adolph[41] reported mean rates of 0.42 and 0.64 gram per min for two subjects of approximately equal surface area in an environment of 76 F, 30 percent RH.

The data revealed no significant difference in the mean rate of total insensible weight loss at the different humidities. This is in agreement with Wiley and Newburgh [17], who reported no differences in evaporative weight loss for a *clothed* subject exposed to relative humidities from 20 to 60 percent, and with Winslow, Herrington and Gagge [31] for nude subjects in the zone of body cooling. Winslow explained this *identical evaporative heat loss* on the basis of an increased wetted area at the higher humidities. For clothed subjects in the zone of body cooling, Winslow, Herrington and Gagge [42] noted that the heat loss by evaporation was only 4 kg (kilogram) cal per (sq m) (hr) less for high than for low humidities. On the other hand, Wiley and Newburgh [17] did find differences in evaporative weight loss at different humidities for a nude subject. Adachi and Ito [43] conducted experiments with one subject on the effect of breathing air at different humidities at 24—25 C. They reported that the loss through the skin varied inversely with the loss from the respiratory passages. Under extremely dry conditions the loss through the skin did not de-

crease sufficiently to compensate for the increased loss through the respiratory passages so that a slight increase in total loss occurred.

In the experiments described in this paper the thermocouple union suits of the subjects were placed in the comfortable room at least 24 hr before use and had attained approximate equilibrium with the environment. In a series of control experiments it was found that for every percent increase in RH, between 30 and 80 percent, the union suit increased approximately one gram in weight. At 80 percent, the union suit put on by the subject, contained approximately 50 g more moisture than when he put the same suit on at 30 percent. Nelbach and Herrington [44] have reported even greater increases in weight in a man's woolen garment for comparable humidity changes. Further, in 4 experiments on one subject at 76 F dry bulb, 2 at 30 and 2 at 80 percent RH, the subject was weighed in the nude, dressed in the union suit and his weight loss followed for one hour. No difference in total weight loss was observed but the union suit had lost more moisture at the higher than at the lower humidity.

TABLE 9. HEAT LOSS BY EVAPORATION ($CAL/M^2/HR$) IN COMFORTABLE ROOM 2

Subject	One Hour Hot Room Exposure			Two Hour Hot Room Exposure		
	RH of Comfortable Room					
	Percent					
	30	60	80	30	60	80
B.F.	33.6	49.1	43.5	66.5	58.2	43.5
J.B.	35.4	62.8	56.9	64.2	56.2	61.9
E.M.	53.5	63.6	52.5	76.6	51.4	56.2
W.P.	59.0	66.5	58.4	76.0	68.4	77.6
M.B.	56.4	68.5	41.9	43.3	70.0	52.7

It is the opinion of the authors that the lack of a significant difference in the average rate of total insensible weight loss under the experimental conditions imposed is due to a number of factors. The fact that the union suits initially contained more moisture at the higher humidities suggests that the measurements included, in addition to the total insensible perspiration, evaporation of moisture made available by the union suit. This also might cause a further compensatory decrease in cutaneous perspiration. The higher mean skin temperature observed at the higher relative humidities, on the other hand, would tend to increase very slightly the cutaneous perspiration. These factors operate to increase the total evaporative weight loss at higher humidities.

Hot Room. On entering the hot room the subject was weighed immediately after the blood pressure and pulse rate were recorded (average time, 3.3 min). The first observed weight averaged 24.3 gram more than the final weight obtained in Comfortable Room 1. This increase was due to the adsorption of moisture on the union suit from the ambient air.

Table 8 shows the evaporative heat loss in calories per square meter per hour for the first and second hours of the two hour hot room experiments. The

values for the one hour hot room experiments have been omitted since they are in the same range as the first hour. In only one experiment did the evaporative heat loss during the first hour exceed that of the second hour, (W.P.) 60 percent. The average evaporative heat loss was 13.87 cal per (sq m) (hr) more ($P = 0.002$) during the second hour. This is accounted for by the delay in diffuse sweating.

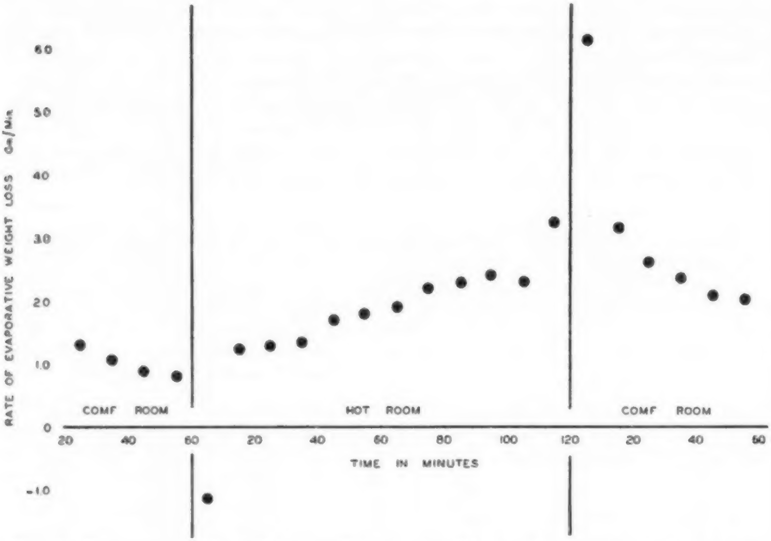


FIG. 8. CHANGES IN MEAN RATE OF EVAPORATIVE WEIGHT LOSS IN GM/MIN FOR ALL SUBJECTS FOR ALL EXPERIMENTS IN A COMFORTABLE ENVIRONMENT, REGARDLESS OF RELATIVE HUMIDITY, FOLLOWED BY EXPOSURE TO A HOT MOIST ENVIRONMENT FOR ONE OR TWO HOURS AND RETURN TO THE COMFORTABLE ENVIRONMENT. EACH POINT REPRESENTS THE MEAN OF 30 EXPERIMENTS, EXCEPT DURING THE SECOND HOUR IN THE HOT ROOM WHERE EACH POINT REPRESENTS THE MEAN OF 15 EXPERIMENTS

Comfortable Room 2. Table 7 revealed that the subjects entered this room with variable amounts of moisture present on the skin and in the union suit. Under the experimental conditions imposed the total evaporative heat loss (Table 9) was apparently unaffected by the RH. On the other hand, if moisture is taken into consideration, it can be demonstrated that the amount of heat loss by evaporation was definitely affected by the relative humidity. By expressing evaporative heat loss in relation to moisture present on the skin and in the union suit at entrance (calories per square meter per hour per gram) the loss was significantly greater ($P = 0.022$) at 30 than at 80 percent RH. No significant difference could be demonstrated for the comparison of 80 *vs* 60 or 60 *vs* 30 percent RH, although the trend was in the same direction as shown.

Fig. 8 shows the mean rate of evaporative weight loss on all subjects for all experiments. In Comfortable Room 1 the mean rate was most rapid immedi-

ately after the subject put on the union suit; in the hot room, after adsorption of moisture was completed (the first point indicates this gain in weight) the mean rate increased with time, and in Comfortable Room 2 the mean rate was most rapid during the first 10 min. Figs. 2 and 3 show the weight changes observed on one subject in two experiments.

Daily Outside Temperatures

Fig. 9 shows that during the period of this study Chicago enjoyed a relatively mild summer. The Annual Climatological Summary, 1946, for Chicago by the Weather Bureau of the United States Department of Commerce stated,

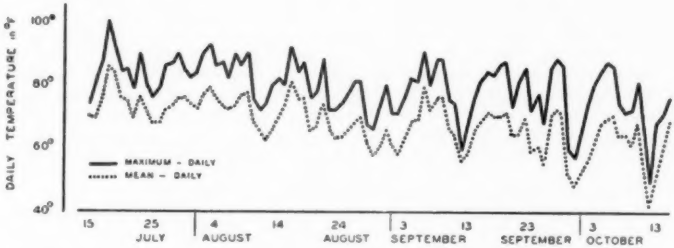


FIG. 9. DAILY MAXIMUM AND MEAN TEMPERATURES FOR CHICAGO AS REPORTED BY THE WEATHER BUREAU

July was warm, dry and sunny, August was cool, dry and sunny, and September was dry and sunny, with average temperature somewhat above normal.

SUMMARY AND CONCLUSIONS

Five healthy male medical students, clothed in 90 percent cotton union suits and wooden sandals, were subjects for six experiments each. The subjects remained in a comfortable room for a control period of one hour. The dry bulb temperature was 76 F for all experiments. The RH was maintained at either 30, 60 or 80 percent. Thus, each subject was exposed to three separate environments. He then went into the hot room (98.5 F dry bulb, 66 percent RH) for either one or two hours. This allowed a variable amount of perspiration to accumulate on the skin and in the union suit. He then returned to the comfortable room and remained for one hour. The subjects remained seated in a Troemner balance in both rooms. Observations on rectal and mean skin temperatures, evaporative weight loss, blood pressure, pulse rate and comfort vote were obtained.

The data justify the following conclusions:

A. Adjustments to exposure of one hour in a comfortable environment at different humidities.

(*a*) The final mean skin and rectal temperatures and pulse rate were significantly higher at the higher humidities; (*b*) the blood pressure was unaltered; (*c*) the

evaporative weight loss varied from one subject to another and from one experiment to another on the same subject so that, under the experimental conditions imposed (greater moisture content of the union suit at the higher humidities), no significant difference was observed at the different humidities; (*d*) the differences in subjective sensations of warmth in the environments of different relative humidity were too slight to be significant and represent differences in individuals. In general, the votes were all 4 or 5.

B. Adjustments to entrance and exposures of one and two hours to a hot moist environment after being in equilibrium in a comfortable environment.

1. On entering the hot environment: (*a*) the mean skin temperature rose quickly and attained approximately its maximum value in less than 10 min; (*b*) the rectal temperature decreased within the first 10 min and then slowly increased; (*c*) the pulse rate and systolic pressure increased, and the diastolic pressure decreased slightly; (*d*) visible perspiration appeared as early as 2½ min and as late as 29 min, but it definitely appeared sooner when the subject had previously been exposed to the comfortable environment with an 80 percent RH as compared to one with 30 percent. In the former environment the subjects were closer to the zone of evaporative regulation.
2. After one hour in the hot room: (*a*) the mean skin temperature was about the same as after 10 min in the hot room; (*b*) the rectal temperature averaged 0.41 F deg higher than the value in the comfortable room; (*c*) the pulse rate was only 2½ beats per minute higher than the initial rate in the hot room; (*d*) the systolic pressure returned to the level observed in the comfortable room while the diastolic pressure remained slightly lower than the final comfortable room value.
3. After an additional hour in the hot room: (*a*) the mean skin temperature was only 0.32 F deg higher; (*b*) the rectal temperature increased 0.55 F deg; (*c*) the pulse rate was 5.1 beats per minute faster; (*d*) the heat loss by evaporation was greater during the second hour than during the first.
4. The comfort votes for two subjects were always 5 (slightly warm) on entrance and on only one occasion, during the second hour, did the vote reach 7 (hot). Three subjects voted *warm* or *hot* on entrance. These three all voted *hot* before the exposure terminated. Differences in individuals were again apparent.

C. Adjustments to entrance and exposure of one hour to a comfortable environment after exposure to a hot moist environment.

1. On entering the comfortable environment: (*a*) the subjective sensation of comfort of most of the subjects was cool (*2*) or cold (*1*). In a few instances a vote of 3 was obtained which seemed to be related to the degree of warmth prior to entrance; (*b*) the rate of decrease in mean skin temperature and the decrease in pulse rate were greater at the lower humidities; (*c*) the rectal temperature increased within the first 10 min and then slowly decreased; (*d*) the blood pressure remained unchanged.
2. The sensation of comfort, or near comfort, was attained earlier at the higher humidities. Some subjects never voted 4 (comfortable).
3. The subjects voted *comfortable* with lower mean skin temperatures than observed in the comfortable room before exposure to heat.
4. The difference in mean skin temperature between a low and a high relative humidity was greater after exposure to heat (Comfortable Room 2) than before exposure to heat (Comfortable Room 1).
5. Although the total evaporative weight loss was not affected by differences in relative humidity the heat loss, expressed in (calories per square meter per hour) (gm) moisture present in the union suit and on the skin at time of entrance, was significantly more at 30 than at 80 percent relative humidity.

In general, the physiological adjustments made by healthy young adult males on passing from a comfortable to a hot moist environment, and vice versa, occurred rapidly and placed little strain on the cardiovascular system. The sub-

jective sensation of warmth is slightly but insignificantly affected by a difference of 50 percent in RH (30-80 percent) when relatively little moisture is present on the skin and in the union suit (Comfortable Room 1). However, after being exposed to a hot moist environment, comfort "4" is attained sooner at the higher relative humidity when large amounts of moisture are present in the union suit (Comfortable Room 2). At 76 F dry bulb, large differences in relative humidity have a slight but significant effect on certain physiological observations.

REFERENCES

1. The Mechanism of Heat Loss in Health and Disease, by E. F. Du Bois. (*Transactions of the Associated Physicians,* 51:252, 1936.)

2. Physiological Reactions of the Human Body to Varying Environmental Temperatures, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*American Journal of Physiology,* 120:1, 1937.)

3. The Relative Influence of Radiation and Convection Upon Vasomotor Temperature Regulation, by L. P. Herrington, C.-E. A. Winslow and A. P. Gagge. (*American Journal of Physiology,* 120:133, 1937.)

4. Basal Metabolism, Radiation, Convection and Vaporization at Temperatures of 22 to 35 C, by J. D. Hardy and E. F. Du Bois. (*Journal of Nutrition,* 15:477, 1938.)

5. A.S.H.V.E. RESEARCH REPORT No. 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, Cyril Tasker and Carl Gutberlet. (A.S.H.V.E. TRANSACTIONS, Vol. 43, 1937, p. 145.)

6. A.S.H.V.E. RESEARCH REPORT No. 1174—Comfort Requirements for Low Humidity Air Conditioning, by F. C. Houghten, H. T. Olson and S. B. Gunst. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, p. 139.)

7. A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337.)

8. A.S.H.V.E RESEARCH REPORT No. 1102—Shock Experiences of 275 Workers after Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 571.)

9. A.S.H.V.E. RESEARCH REPORT No. 1103—General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley and E. Witkowski. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 591.)

10. The Role of the Extremities in the Dissipation of Heat from the Body in Various Atmospheric and Physiological Conditions, by Charles Sheard, M. M. D. Williams, G. M. Roth and B. T. Horton. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 135.)

11. Thermoregulatory Phenomena Associated with Exposure to Warm and Cold Environments, by J. D. Hardy and H. Goodell. (*Proceedings of the American Physiological Society,* 1947.)

12. The Tolerance of Man to Cold as Affected by Dietary Modifications: Protein Versus Carbohydrates, and the Effect of Variable Protective Clothing, by R. W. Keeton, E. H. Lambert, Nathaniel Glickman, H. H. Mitchell, J. H. Last and M. K. Fahnestock. (*American Journal of Physiology,* 146:66, 1946.)

13. The Technique of Measuring Radiation and Convection, by J. D. Hardy and E. F. Du Bois. (*Journal of Nutrition,* 15:461, 1938.)

14. The Probable Error of a Mean. *Student.* (Biometrika 6:1, 1908.)

15. New Tables for Testing the Significance of Observations. *Student.* (Metron 5:105, 1925.)

16. The Effects of High Humidity on Skin Temperature at Cool and Warm Conditions, by H. Freeman and B. A. Lengyel. (*Journal of Nutrition,* 17:43, 1939.)

17. The Relationship Between the Environment and the Basal Insensible Loss of Weight, by F. H. Wiley and L. H. Newburgh. (*Journal of Clinical Investigation,* 10:689, 1931.)

18. Humidity, Skin Temperature and Sense of Comfort, by U. Miura. (*American Journal of Hygiene,* 13:457, 1931.)

19. Skin Temperatures of Children, by F. B. Talbot. (*American Journal of Diseases of Children,* 42:965, 1931.)

20. Studies in Ventilation: Skin Temperature as Related to Atmospheric Temperature and Humidity, by E. B. Phelps and A. Vold. (*American Journal of Public Health,* 24:959, 1934.)

21. Physiological Reactions of the Human Body to Various Atmospheric Humidities, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*American Journal of Physiology,* 120:288, 1937.)

22. Experimentelle Untersuchungen über die tierische Wärme, by K. Yoshinaga. (*Mitteilungen aus der Medizinischen Fakultat, Imperial University, Fukuoka.,* 10:161, 1925.)

23. Zur Lehre von der Wärmeregulation, by W. Filehne. (*Archiv für Physiologie,* 1910, p. 551.)

24. Über den Schweiss und das Schwitzen, by A. Strasser. (*Zeitschrift für Physikalische und diatetische Therapie,* 18:129, 1914.)

25. The Physiology of Human Perspiration, by Y. Kuno. (Churchill, London, 1934, 268 pp.)

26. On the Change in the Rectal Temperature of Man When His Body Is Exposed to Extremely Cold Atmosphere, by R. Shoji and K. Ogata. (*Japanese Journal of Medical Science,* Biophysics, 6:85, 1939.)

27. The Measurement of Skin Temperature in Its Relation to the Sensation of Comfort, by E. F. Ward. (*American Journal of Hygiene,* 12:130, 1930.)

28. Relation Between Atmospheric Conditions, Physiological Reactions and Sensations of Pleasantness, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*American Journal of Hygiene,* 26:103, 1937.)

29. The Influence of Clothing on the Physiological Reactions of the Human Body to Varying Environmental Temperatures, by A. P. Gagge, C.-E. A. Winslow and L. P. Herrington. (*American Journal of Physiology,* 124:30, 1938.)

30. Heat Loss from the Nude Body and the Peripheral Blood Flow at Temperatures of 22 C to 35 C, by J. D. Hardy and G. F. Soderstrom. (*Journal of Nutrition,* 16:493, 1938.)

31. The Importance of Clothing in Air Conditioning, by C. P. Yaglou and A. Messer. (*Journal of the American Medical Association,* 117:1261, 1941.)

32. A.S.H.V.E. RESEARCH REPORT No. 1108—Cardiac Output, Peripheral Blood Flow, and Blood Volume Changes in Normal Individuals Subjected to Varying Environmental Temperatures, by F. K. Hick, R. W. Keeton, Nathaniel Glickman and H. C. Wall. (A.S.H.V.E. TRANSACTIONS, Vol. 45, 1939, p. 123.)

33. A.S.H.V.E. RESEARCH REPORT No. 1151—The Peripheral Type of Circulatory Failure in Experimental Heat Exhaustion. The Role of Posture, by R. W. Keeton, F. K. Hick, Nathaniel Glickman and M. M. Montgomery. (A.S.H.V.E. TRANSACTIONS, Vol. 46, 1940, p. 157.)

34. The Influence of High Air Temperatures, by J. S. Haldane. (*Journal of Hygiene,* 5:494, 1905.)

35. A.S.H.V.E. RESEARCH REPORT No. 654—Some Physiological Reactions to High Temperatures and Humidities, by W. J. McConnell and F. C. Houghten. (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 129.)

36. A.S.H.V.E. RESEARCH REPORT No. 672—Further Study of Physiological Reactions, by W. J. McConnell, F. C. Houghten and F. M. Phillips. (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 353.)

37. The Index of Comfort at High Atmospheric Temperatures, by H. M. Vernon. (*Medical Research Council Special Reports,* No. 73:116, 1923.)

38. Effects of Exposure to High Temperatures Upon the Circulation in Man, by E. F. Adolph and W. B. Fulton. (*American Journal of Physiology,* 67:573, 1924.)

39. Evaporation from Human Skin with Sweat Glands Inactivated, by Ernest A. Pinson. (*American Journal of Physiology,* 137:492, 1942.)

40. Insensible Perspiration: Its Relation to Human Physiology and Pathology, by F. G. Benedict and H. F. Root. (*Archives of Internal Medicine,* 38:1, 1926.)

41. The Initiation of Sweating in Response to Heat, by E. F. Adolph. (*American Journal of Physiology,* 145:710, 1946.)

42. The Reactions of the Clothed Human Body to Variations in Atmospheric Humidity, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*American Journal of Physiology,* 124:692, 1938.)

43. Observations on the Insensible Loss from the Skin and the Respiratory Passages, by J. Adachi and S. Ito. (*Journal of Oriental Medicine,* 21:103, 1934.)

44. A note on the Hygroscopic Properties of Clothing in Relation to Human Heat Loss, by J. H. Nelbach and L. P. Herrington. (*Science,* 95:387, 1942.)

DISCUSSION

E. F. DuBois, M.D., New York, N. Y. (Written): The main point of this paper is that large differences in relative humidity have a slight but significant effect at 76 F dry bulb. In practice it would seem advisable for heating and ventilating engineers to determine whether it is more practicable to control the relative humidity or the dry bulb or the air movement in order to obtain better comfort. It must be remembered that these experiments are performed on young healthy males and that results might not apply to women or older subjects. The results seem to point to the necessity for more studies of surface temperature as an index of comfort. It would be advisable to make an extensive survey of skin temperatures of rather large groups of people under actual working conditions.

The results of these and other studies sponsored by the A.S.H.V.E. are of considerable importance to physiologists and they should be published from time to time in physiological journals.

J. D. Hardy, M.D., New York, N. Y. (Written): I have read over the paper by Glickman et al and have the following comments to make:

1. The work is a step forward in the right direction in that it is a study of changes that occur in the body in response to a changed thermal environment.

2. More care is given the matter of relative humidity all the way through from the cool to the warm environment.

3. Much more attention has been paid to skin temperature as the important organ of heat loss.

4. The conditions which were studied most carefully were those under equilibrium circumstances, that is, after an hour or more exposure to allow transient responses to disappear.

In keeping with the past excellent work of these writers, they are studying comfort from the verbal report of the sensations of the subjects. This is entirely correct and I feel there is no other way to approach the problem. However, as thermal sensations are related to *changes* in skin temperature rather than to the *level* of the skin temperature, I think much more attention should be focused on such changes as related to comfort. Yaglou has proposed the actual level of the skin temperature as a dependable index correlated, to a high degree, with comfort. I feel that a level skin tem-

perature and rectal temperature indicate only that the body is in a temperature zone in which it can regulate physiologically. If, however, the skin temperature has to go through cyclic changes about a mean, such as we have observed in the sweating and shivering zones, strong sensations arise from the periphery and cause discomfort. I feel that we, as well as others, shall probably find that comfort is the zone in which these changes are minimal.

AUTHORS' CLOSURE: We are in accord with the comments of Dr. DuBois. These experiments are now being repeated with older subjects, and it is planned to obtain similar experiments with women.

It would, indeed, be advisable to perform studies similar to those of Dr. Bedford on large groups of people under actual working conditions and attempt to correlate comfort with mean skin temperatures. Perhaps we shall be able to do this in the future.

Referring to Dr. Hardy's comments, when subjects are in equilibrium with the environment, the level of mean skin temperature is a good index of the degree of warmth experienced. However, when the environmental conditions are changed suddenly or the subject is not in equilibrium with the environment, the level of mean skin temperature is a poor index of the degree of warmth experienced. Under these latter conditions, the change of mean skin temperature may give a better index, although this alone probably is not the best index.

We have been attempting to analyze our data to determine the relationship between sensations of warmth and change of mean skin temperature at different levels of mean skin temperature. As Dr. Hardy mentions, comfort is in the zone where these changes are minimal, but this does not obtain when the mean skin temperatures are above or below the range indicated for comfort. If the subject is in an environment which is above the comfort zone, then the end organs for the perception of heat are stimulated, although the changes in the mean skin temperature may be minimal. A similar condition with respect to cold sensations may exist in an environment below the zone of comfort.

1321

No. 1321

COMFORT REACTIONS OF WORKERS DURING OCCUPANCY OF AIR CONDITIONED OFFICES

By FRANK B. ROWLEY*, RICHARD C. JORDAN**, AND
WARREN E. SNYDER†, MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, in cooperation with the Engineering Experiment Station, University of Minnesota.

THE effective temperature scale [1] which relates the dry bulb temperature and relative humidity to the feeling of warmth was derived largely through A.S.H.V.E. Laboratory investigations. Inasmuch as the scale is to be applied to air conditioning installations where a number of additional comfort affecting factors, such as solar radiation, health of the individuals, attitude toward the test, etc., are present, a check of its validity during practical application was desired.

Suitable data for this study were available from a cooperative survey previously made in the air conditioned offices of the Minneapolis-Honeywell Regulator Co. These offices were occupied by 275 employees, including both men and women, who had an age distribution of from slightly below 20 years to 70 years and who were, in general, representative of persons found in a typical office. The data were used originally to determine the optimum effective temperature for summer cooling requirements in Minneapolis and were reported in the A.S.H.V.E. TRANSACTIONS [2,3,4].

METHOD OF OBTAINING DATA

The original data were obtained from questions answered by the employees. One question, namely, general reactions during working hours, was independent

*Head, Department of Mechanical Engineering, and Director, Engineering Experiment Station, University of Minnesota. Member of A.S.H.V.E.
**Professor and Assistant Head, Department of Mechanical Engineering, University of Minnesota. Member of A.S.H.V.E.
†Instructor, Department of Mechanical Engineering, University of Minnesota.
[1]Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Coronado, Calif., June 1947.

of entering or leaving shock effects and was the only question used in the survey. Since this information was obtained at approximately 10:30 a.m. and 3:00 p.m., the subjects had arrived at a steady physiological state and were adapted to the room temperature. In the development of the original ET scale, the subjects recorded their reactions when passing from one atmospheric condition to another. Therefore, it is not to be expected that the steady state reactions will necessarily be the same as those obtained by the comparison method,

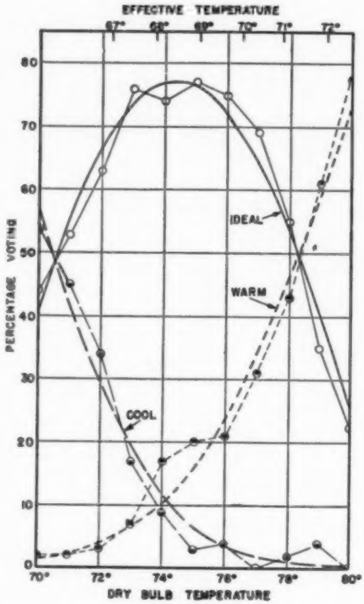


FIG. 1. 40 PERCENT RH

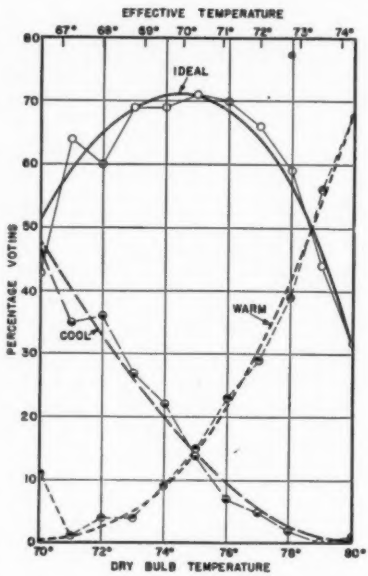


FIG. 2. 55 PERCENT RH

and the question arises as to which should be used as the basis for effective temperature.

The A.S.H.V.E. Research Laboratory has used over a period of years a comfort reaction scale in which *(1)* represented severely cool; *(2)* cool; *(3)* comfortably cool; *(4)* ideal comfort; *(5)* comfortably warm; *(6)* warm; and *(7)* severely warm. In recording the original data the classifications of comfortably cool *(3)* and comfortably warm *(5)* were found to be confusing and, thus, were eliminated. The reactions checked were severely cool *(1)*, mildly cool *(2)*, ideal *(4)*, mildly warm *(6)*, and severely warm *(7)*. In making the present study it was found that the percentages of persons voting severely cool *(1)* and severely warm *(7)* were quite small and erratic. The tabulation

was further simplified by combining all cool votes and all warm votes, leaving only the three classifications of cool (*2*), ideally comfortable (*4*), and warm (*6*).

TABULATION OF DATA

In the present study all pertinent information for each questionnaire, including the date, name, sex, age, location in building, inside temperature and humidity, outside temperature and humidity, and answers to all questions, was coded and punched on an IBM card. These cards were first sorted by inside dry bulb temperatures in intervals of 1 deg and by inside relative humidity in intervals of 5 percent. For the purpose of this grouping, humidities from 33 to 37 percent inclusive were combined and called 35 percent, humidities from 38 to 42 percent inclusive were combined and called 40 percent, etc. The total number of votes recorded for cool, ideally comfortable and warm was tabulated for each condition and the percentage of cool, ideally comfortable and warm votes was calculated. It was found that the number of votes contained in the extremes of the data was too small to be significant and these votes were discarded, leaving the temperature range from 70 F to 80 F and the humidity range from 35 to 60 percent.

Six graphs were constructed, one for each 5 percent group of relative humidities from 35 to 60 percent. Two typical graphs are shown in Figs. 1 and 2. The dry bulb temperature was used as the abscissa and the percentage of persons voting for each of the three conditions (*cool, ideally comfortable,* and *warm*) as the ordinate. After the points were plotted, smooth curves were drawn through them and the remainder of the study was based on data from these smoothed curves rather than on the original data. This was done in order to minimize the dispersion of points and to obtain percentages corresponding to effective temperatures.

Using the values as obtained from the six graphs, new graphs were plotted for each effective temperature from 67 to 72 deg, with the percentage of persons voting for each of the three conditions as the ordinate and with the relative humidity as the abscissa. Three typical curves are shown in Figs. 3, 4 and 5 for 67, 70 and 72 deg ET respectively.

Again using values as obtained from the original six graphs, additional graphs were made for dry bulb temperatures from 70 F to 80 F, with the percentage of employees voting for each of the three conditions as the ordinate and the relative humidity as the abscissa. Four typical curves are shown in Figs. 6 through 9 for temperatures of 70, 74, 78 and 80 F.

A study of Figs. 3, 4 and 5 showing the variation in the percentages of employees feeling cool, ideally comfortable and warm for a given effective temperature, reveals that in every graph the percentage of occupants feeling warm decreases and the percentage feeling cool increases as the humidity increases with corresponding decreases in the dry bulb temperature, even though the effective temperature remains fixed. These changes in the percentages would not be expected at a constant effective temperature; for, by definition, an effective temperature is one which indicates conditions of equal warmth. The fact that these curves do show that the employees studied became cooler as the humidity increased and dry bulb temperature decreased, despite a constant effective temperature, would indicate that with this particular group of employees,

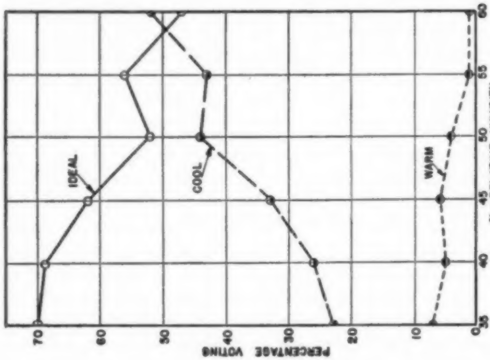


FIG. 3. 67 DEG ET

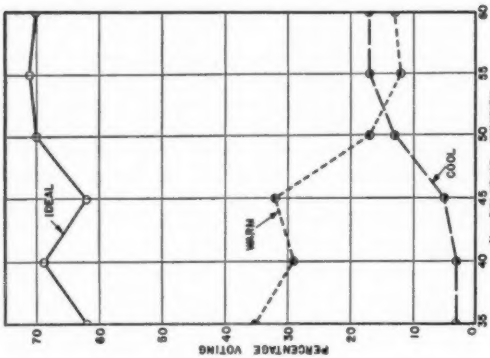


FIG. 4. 70 DEG ET

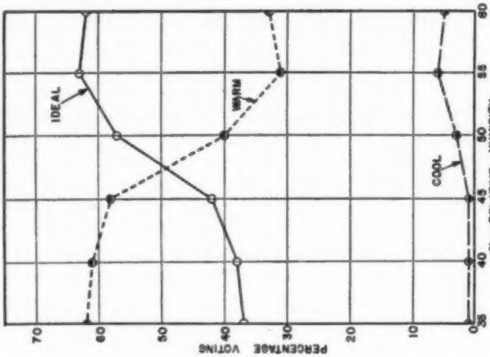


FIG. 5. 72 DEG ET

humidity has less effect and dry bulb temperature more effect on their feeling of comfort than is shown by the effective temperature relationship.

COMFORT NOT AFFECTED BY HUMIDITY CHANGE

A study of Figs. 6 through 9 showing the variation of percentages of the employees feeling cool, ideally comfortable and warm for a constant dry bulb temperature indicates that there is no appreciable change in the percentages as the relative humidity increases. This indicates that within the limited range of relative humidities investigated, 35 to 60 percent, the employees studied felt

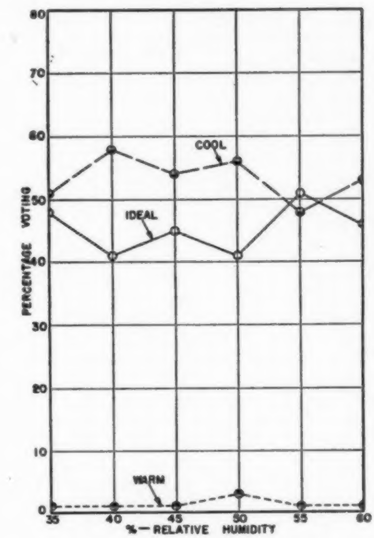


FIG. 6. 70 DEG DRY BULB TEMPERATURE

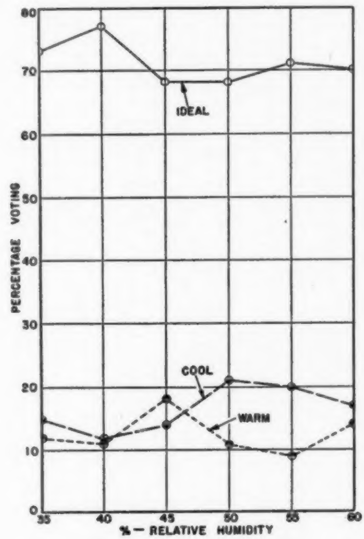


FIG. 7. 74 DEG DRY BULB TEMPERATURE

no appreciable change in comfort with a change of humidity at a constant dry bulb temperature.

The seeming lack of any humidity effect on the feeling of warmth is in direct contradiction to the effective temperature concept and could be ascribed partly to inherent limitations of the ET index and partly to the many factors entering into an individual's feeling of warmth, factors which could be controlled in the laboratory but not in a business office. Some of these factors are solar radiation, health of the individual, drafts, attitude toward the test, and variation in the amount of clothing worn from day to day.

In order to check the effects of these factors, several charts were plotted, each recording all of the votes cast by various persons who were chosen at random.

These charts were plotted with relative humidity as ordinate and with dry bulb temperature as abscissa. One typical chart is shown in Fig. 10. The three diagonal lines represent effective temperatures of 66, 71 and 75 deg, which are given in the ET scale as the minimum, optimum, and maximum effective temperatures for summer comfort. These figures show that there was a wide variation in the attitude and reactions as obtained from the individual records.

The individual whose votes are shown in Fig. 10 was a woman under 20 years of age. The record of her daily votes indicates little consistency, showing ideal votes throughout the range of conditions, and also cool and warm votes scattered seemingly indiscriminately throughout most of the range. Further study of this individual failed to show any ascertainable correlation between her feeling of warmth and other factors except that the votes cast during the morning were slightly more consistent than those cast in the afternoon. It was thought that possibly one factor here might be the effect of solar radiation, inasmuch as this person was located in the northwest corner of the building directly inside the western exposure windows.

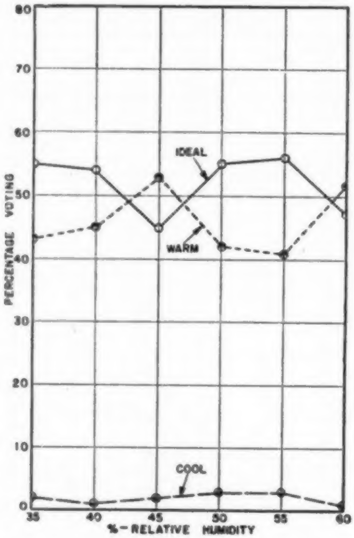


FIG. 8. 78 DEG DRY BULB TEMPERATURE

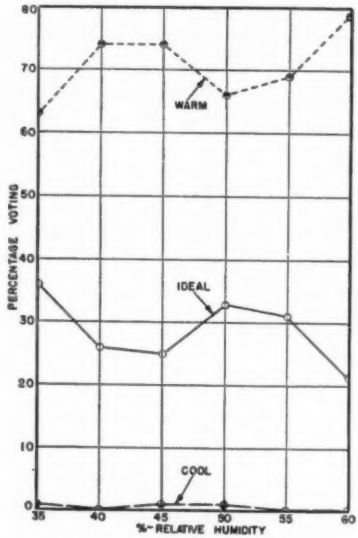


FIG. 9. 80 DEG DRY BULB TEMPERATURE

Taken together, the charts of individual votes show clearly that there was little correlation in the feelings of warmth between one person and another and, in some cases, little correlation between the feeling of warmth and condition of temperature and relative humidity for one person.

A further question arose during the study of individual reactions when it was found that the total number of votes varied widely from person to person, ranging from eight to ninety among the individuals studied.

The effect of this variation could not be ascertained, but whether due to incomplete polling or to some other factor, it could possibly cause serious errors in the results obtained.

Based on an analysis of the reactions of selected individuals, the graphs showing the average reactions of the entire group take on a new meaning. As averages, they reflect the conglomerate effect of individuals of different temperament and varying degrees of consistency and cooperation. While this study indicates that there was no apparent effect of humidity on comfort for this office and group of employees, the question arises as to whether or not the same relationship would hold true in another office with a different group of employees.

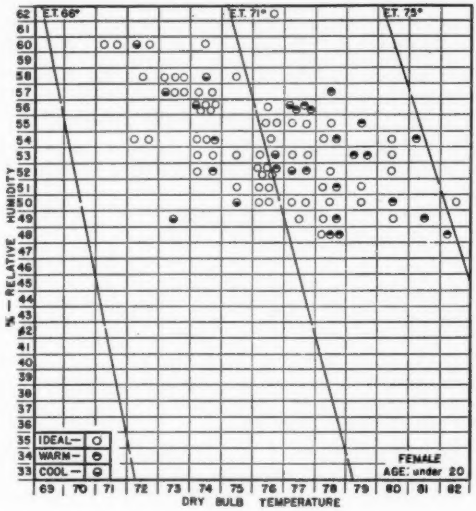


FIG. 10. PERSON "G"

CONCLUSIONS

With these limitations emphasized, the general conclusions to be drawn from this study of effective temperature are as follows:

1. For this office and particular group of employees and in the limited range of conditions studied, variations of RH between 35 and 60 percent caused no appreciable effect in the feeling of comfort.

2. The maximum percentage of subjects indicating comfort occurred at a dry bulb temperature of about 74 F regardless of humidity (68.5 to 70.5 deg ET). Seventy percent of the employees were included in this group.

3. The lack of any humidity effect on the feeling of warmth is in direct contradiction to the effective temperature concept and could be ascribed partly to inherent limitations of the ET index and partly to the many factors entering into an individual's feeling of warmth which could be controlled in the laboratory, but not in a business office.

4. It is recommended that laboratory experiments be undertaken for the purpose of confirming or discrediting the disputed humidity effect within the zones of comfort and for lower temperatures.

It is emphasized that the results of this study are based upon a limited range of temperature and humidity and that a considerable period of time elapsed between the taking of the data and the analysis of the data. Furthermore, the original data were not taken by the authors and were not taken with the present analysis specifically in view. While it does appear that the effective temperature scale needs revision, it is not recommended that this report be used as the basis for that revision. It is felt that the results presented herein are not in themselves conclusive evidence and should not be taken as other than indicative of some need for a revision based upon more complete information.

REFERENCES

1. HEATING, VENTILATING, AIR CONDITIONING GUIDE 1947, p. 217.

2. A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 337.)

3. A.S.H.V.E. RESEARCH REPORT No. 1102—Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 571.)

4. A.S.H.V.E. RESEARCH REPORT No. 1103—General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 591.)

DISCUSSION

F. C. McINTOSH, Pittsburgh, Pa. (WRITTEN): Although the data presented by the authors are too few to justify final answers, there seem to be enough to indicate definite disagreement with our Comfort Chart.

I have a special interest in this subject. About three years ago, I started preparing a paper for the Society on the merits of controlling indoor temperatures higher in cold weather than in moderate weather. This applied to the heating season. The greatest of three factors was the humidity effect shown in the A.S.H.V.E. Comfort Chart. In the Pittsburgh district, this indicated that the dry bulb temperatures should be varied over a range of about 4 deg, as the relative humidity changed between 25 and 70 percent.

In order to have more material for the paper, I arranged for field tests in three buildings having a total of about 40 rooms under control. Thermostats were set for automatic temperature increase in colder weather (by humidity effect). In a short time I found that something was wrong with the calculations because complaints of being too warm came in dry rooms. Eventually the correction was reduced to one half the initial amount, and it is my opinion that this correction is taking care of the factors, other than humidity, which were included in the calculation. This experience

may have been due to the negligible importance of relative humidity in the winter comfort zone.

At the same time I started reading and inquiring on the subject. Dr. C.-E. A. Winslow had evidence that disagreed with the Comfort Chart. Prof. C. P. Yaglou, who worked on the original society project,[5] no longer accepted those findings. Even the HEATING, VENTILATING, AIR CONDITIONING GUIDE has referred to other studies indicating that the Comfort Chart may not show the true effect of relative humidities.

The Society has used this chart for nearly 25 years with no changes, and criticism of it seems to have been treated as heretical. In my opinion we owe the authors a vote of thanks for presenting their conclusions so clearly and we should follow the recommendation they have made.

C. S. LEOPOLD, Philadelphia, Pa. (WRITTEN): The report as originally submitted contained a number of graphs which are not included in the final report. In this discussion it is necessary to make some reference to these findings.

According to the present ET lines, the entire variation from 35 to 60 percent relative humidity is approximately 3 deg F dry bulb. The general effect of relative humidity is accepted by practically all serious investigators and the question is only one of the degree of this effect. Since the analysis here presented showed an actual reverse in the effect of relative humidity, it is advisable that we check the experimental procedure to see if these inconsistent effects could be due to a systematic error in procedure.

According to the original report,[6] the range of acclimatization over the period of the test was from 69 to 74 ET which, at constant humidity, is approximately 8 deg dry bulb. Since, for the present report, results for the entire period were considered, any systematic error introduced by acclimatization is also included and a small systematic error in an 8 deg effect would vitiate conclusions drawn as to a change in a 3 deg effect. To check within 3 deg dry bulb would require an experimental set-up capable of an accuracy of less than 3 deg.

Such accuracy implies the complete elimination of all variables except those being investigated and a mathematically adequate distribution of maintained conditions. There is some evidence that the test did comply with these qualifications.

Although this is the fourth report based on these experiments, the log of daily operating conditions has not been made available. It is, therefore, necessary to deduce a portion of the experimental background from the previously published paper.

Test Procedure: From the earlier report[7] we learn that three sets of apparatus were used to supply the test area. All apparatus utilized 52 F well water for dehumidification and cooling. Apparatus No. 1 employed an air washer and the other two utilized *dry* coils.

1. In order to maintain satisfactory dew-points, the average commercial job requires water approximately 6 deg cooler than the well water used in the experiments.

2. If reheat were not used in warm and hot weather, as has been verbally reported, the ability to maintain relatively low humidities would be further affected.

3. Since the operating log is not available, it is necessary to refer to the log sheets for individuals (included in the full report). Collectively they indicate that there were few tests made at humidities below 45 percent.

4. Inspection of the data for conditions maintained for person G, a.m. and p.m. separately recorded, indicates that in general the afternoon conditions were warmer and drier, with higher effective temperatures, than the morning.

5. A condition reported as *warm* by an individual acclimated to 69 F could be reported as *cool* to the same individual acclimated to 73 F, and this error could only be eliminated if there were an even distribution of observations for each combination of relative humidity and dry bulb for each effective temperature to which the subjects became acclimated over the period of the tests. If, as appears from the data, high

[5]Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou. (A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 361.)

[6]A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 347.)

[7]A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 341.)

indoor humidities tended to accompany high outdoor temperatures, and low indoor humidities tended to accompany low outdoor temperatures, the error would be of sufficient magnitude to nullify the conclusions.

As a result of the limitations in the means of dehumidification (1 and 2), cool conditions could only be obtained in warm weather by producing high humidities and low temperature. Similarly, low humidities could only be obtained when the outside dew-point was low, particularly for Apparatus No. 1.

Inspection of the graphs for individuals (3, 4, and 5) presents additional evidence that the distribution of conditions maintained was inadequate for a statistical analysis.

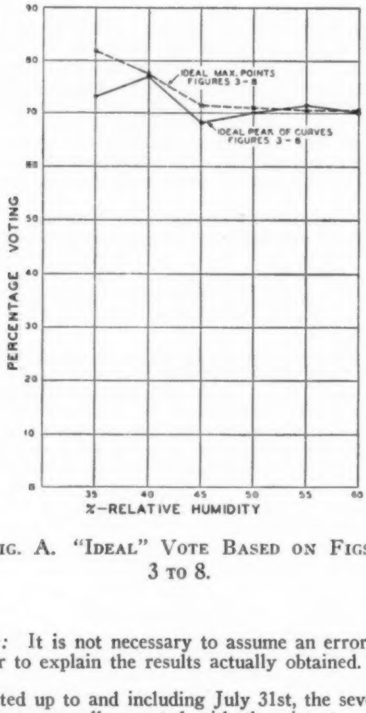


FIG. A. "IDEAL" VOTE BASED ON FIGS. 3 TO 8.

Actual Test Results: It is not necessary to assume an error in the effective temperature lines in order to explain the results actually obtained. The original report[6] states:

"For the data collected up to and including July 31st, the several zones conditioned by the different systems were all operated with the view to giving maximum comfort without reference to the study. . . .

For the period August 1 to September 30, occupants in the different zones, or at different times within the same zone, were given a range of effective temperatures designed to give a greater spread when plotted as in Fig. 2. The variations in effective temperature for optimum comfort for this latter period in the summer are therefore somewhat more significant than for the earlier period."

For the period May through July, we would expect a relatively constant line for *ideal,* as shown in Fig. 7, as this was exactly the condition the operators were trying to produce and they would, therefore, be expected to take into account acclimatization. Because of the test procedure, it would appear that as a minimum the analysis should deal separately with the tests before and after July 31.

The graphs for constant temperatures indicate, in many cases, that increased RH increased the votes for *cool,* which is not only at variance with the effective tem-

perature lines but is even at variance with those investigators who question the degree to which an increase in relative humidity tends to increase the feeling of warmth. The results, however, can be readily explained by realizing that with the subject apparatus, and in hot weather, moderate or low effective temperatures could be obtained only by the combination of low dry bulb and high humidity.

Consideration of this limitation on the production of test conditions, together with the effect of acclimatization, would indicate that Fig. 5 is the reasonable outcome of the experimental procedure, even if the effective temperature lines are correct to the minutest detail.

Were the experimental data valid, we could conclude that 35 to 40 percent RH would satisfy more people than any other. This is shown in Fig. A (which I herewith submit and immediately disown as a generality) where the dots illustrate points selected as the peak of the smooth curves and X equals the highest points for Figs. 3 to 8. For the smooth curve there is a peak in *ideal* at 40 percent with a drop at 45 percent then a partial recovery. For points, the maximum is at 35 percent with a sharp drop at 45 percent. As generalizations, both results appear improbable and the error probably rests in peculiarities of the installation or its operation. For example, at humidities of approximately 45 percent and below, well water could not be used in the air washer of Apparatus No. 1 if it were desired to produce low effective temperatures without unduly increasing humidity and the points of use or non-use would

TABLE A—COMFORT TEMPERATURES

ALL MEN AND WOMEN	OPTIMUM ET	IDEAL MAXIMUM PERCENT
May 20 to June 10	69	57
July	71.5	55
August	70.	75
September	68.5	67

introduce a variable. Apparently this system performed better when not called upon for dehumidification.

Referring again to the original report,[8] Fig. 7, we obtained the results shown in Table A.

These results are significant in that the percentage of people satisfied is considerably lower than that reported in similar tests in other cities and this is particularly true of the period May 20 to June 10 and for the month of July, at a time when the operators were trying to produce maximum comfort.

Assuming the adequacy of the method of sampling opinion, these results would indicate limitations of the apparatus or its operation. In the absence of the operating log, we have no means of telling what index the operators used to establish comfort and whether or not they were trying to follow the outdoor temperature.

In order to be able to draw any conclusions on the effect of relative humidity, it would be necessary to break down the data as, for example, on the basis of a day or less or, at the most, on the middle portion of a hot or cool spell, avoiding the first day. Interesting information might also be obtained by separately analyzing the areas served by Apparatus No. 1 as compared with Nos. 2 and 3.

Conclusion: In view of the foregoing, I hope that the authors agree that their report should be critically re-examined because, unless my comments on acclimatization and the distribution of observed conditions are invalid, there is no basis for drawing any conclusion as to the effect of relative humidity on effective temperature.

The authors suggest that the lack of any humidity effect on the feeling of warmth might be due to factors entering into an individual's feeling of warmth, which could be controlled in a laboratory but not in a business office. I believe that the real

[8]A.S.H.V.E. RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (A.S.H.V.E. TRANSACTIONS, Vol. 44, 1938, p. 349.)

difficulty was not due to the particular group of people or the physical arrangement of the office but was largely due to the lack of flexibility of the apparatus to provide conditions to cover a full range, as required for a test of this type. In other words, the experimental procedure does not justify the statistical analysis.

I agree with the authors' recommendation of further tests on effective temperature. Before undertaking these tests, however, I think a critical review of previous work is in order and that the first experimental work should be devoted to the re-establishment of the winter optimum effective temperature line, which has been shown by the later work of the Laboratory to be at important practical variance with the 1923 experiments. It is further suggested that in conducting these tests the Laboratory equip its apparatus with electrostatic filters and charcoal adsorbers in order to minimize the odor effects which frequently accompany high humidities.

R. S. L. ARNOLD, Philadelphia, Pa., (WRITTEN): The conclusions drawn by the authors undoubtedly indicate that more investigation along this line should be made. Before we discount too much the reliability of applying the ET chart to air conditioning installations, it would certainly seem that other studies should be made taking into consideration a number of factors which have not been mentioned in this paper. It is noted that no mention has been made regarding the conditions of air motion under which these tests were made. Air motion, as well as air distribution, has a profound effect on sensations of comfort.

Furthermore, radiant heating and radiant cooling have considerable effect on comfort and this factor should be taken into consideration. It is well recognized that occupants of a room who are located near an exposed wall would experience a large amount of radiation effect on severe days whereas in this same room the occupants near inside walls would experience very little effect from radiation.

Other factors enter into a person's feeling of comfort, such as rate of metabolism and state of mind at the time when subjected to a certain condition of atmosphere within the space occupied.

No doubt this study has a great deal of merit but more investigation should be made before stating too definite a conclusion regarding the reliability of the ET chart.

AUTHORS' CLOSURE: The authors are in agreement with some of the detailed criticism offered, but are in disagreement with other comments. However, we feel that a complete, detailed discussion of each point would be of doubtful value as the prime objectives might then be placed out of focus.

We welcome the opportunity of reiterating our position in these researches. First, we are unable to take either credit or blame for the initial organization and procedure of the project. As stated in our paper, our analyses were restricted to the framework already established. Second, comfort reaction research must, of necessity, be statistical and it was therefore necessary to assume, as an initial premise, the validity of the data obtained by Dr. F. C. Houghten, A. B. Newton, et al. Third, the analyses made on this basis yielded results in conflict with established concepts of effective temperature and comfort reactions, because of either (1) limitations in the present ET scale, or (2) limitations in the data. Fourth, because we do not feel that further analyses of the present data will result in any more definite or uncontestable conclusions and because the range of the data is insufficient to establish any revision of the scale, we have recommended further laboratory experimentation as an initial step. If this contribution acts as a catalyst toward undertaking such investigations, our objectives will have been accomplished.

No. 1322

EXPERIMENTAL STUDIES ON PANEL HEATING TUBE SPACING

By B. F. Raber*, Berkeley, Calif. and F. W. Hutchinson**, Lafayette, Ind.

In previous panel heating research publications [1-5] the authors have confined their attention to the influence on thermal environment of a uniformly heated flat surface located in the ceiling, or floor, or wall. No consideration was given to the mechanism by means of which the surface temperature could be raised to the required value nor to the problem of achieving uniformity of temperature across the surface. This paper differs from the others in that it is concerned solely with conditions behind the heating surface and presents results of a detailed experimental investigation of various sizes and spacings of copper tubing when embedded in plaster and used to convey hot water.

Problem

As elsewhere described [1-5], the first problem in the design of a panel heating system is to determine the required panel area for a selected design surface temperature, which will permit carrying maximum heating load. Irrespective of the type of panel to be used—whether energized by electricity, warm air, steam, or hot water—the area and temperature can be fixed in terms of the thermal characteristics of the structure. When a heat balance analysis has been completed it is then possible, from a knowledge of the equilibrium optimum room air temperature and the average temperature of unheated room surfaces, to evaluate the required energy dissipation rate, expressed in Btu per (square foot of panel) (hour), of a panel surface operating at a fixed temperature. Until a heat balance has been established the panel output *cannot* be determined since it depends on two variables—the inside air temperature and the average temperature of unheated room surfaces—which have optimum comfort values that

*Professor of Mechanical Engineering, University of California, Member of A.S.H.V.E.
**Professor of Mechanical Engineering, Purdue University. Member of A.S.H.V.E.
[1]Exponent numerals refer to References.
Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Coronado, Calif., June 1947.

cannot be fixed in advance of analysis. Even in cases where the ventilation rate is so low that the optimum comfort air temperature does not vary greatly with the type of structure, the unheated surface temperature will be found to change so much that variations in panel output—for fixed panel surface temperature and fixed air temperature—will frequently be of the order of 15 percent.

Consider, for example, an 85 F floor panel of area sufficient to maintain comfort in a structure with optimum inside air temperature of 69 F when outside

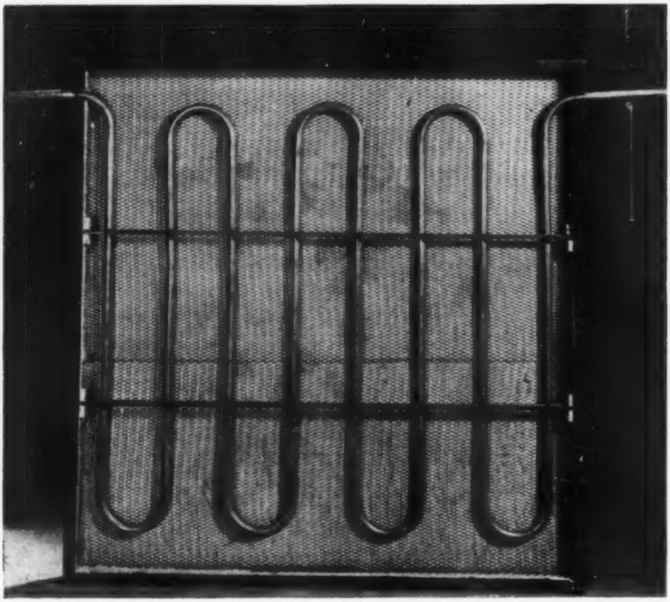


FIG. 1. REAR VIEW OF REPRESENTATIVE TEST PANEL PRIOR TO APPLICATION OF PLASTER

design temperature is +30 F and ventilation is at a rate of 1 cfh per square foot of room surface. If the equivalent overall heat transmission coefficient for the structure were 0.04 the heat balance analysis [8] would indicate a panel output of 34 Btu per (hr) (sq ft of panel) with an exact optimum inside air temperature a fraction of a degree above 69 F. If, on the other hand, the equivalent overall coefficient were 0.26, the analysis would show a panel output of 39 Btu per (hr) (sq ft) with an exact optimum inside air temperature of a fraction of a degree below 69 F. Thus for these two structures unit area of panel operating with fixed surface-to-air temperature difference of 85 — 69 = 16 would have outputs differing by approximately 14 percent. If, in such an instance, the design were based on the assumption that the rating of the panel in a comfort

installation is fixed in terms of the panel surface temperature the resultant error in determination of requisite panel area might be as much as 14 percent.

The above analysis indicates that an "overall" correlation of panel ratings in terms of water-to-air or surface-to-air temperature differences is irrational and unlikely to lead to dependable coefficients. A more striking demonstration of the same fact is that a floor panel operating at a fixed temperature dissipates 50 percent more energy than a ceiling panel operating at the same surface temperature. Hence it is deduced that an analysis of the thermal environment in a comfortable room must end at the surface of the panel, whereas analysis of the mechanism of bringing energy from the source (as water in a tube) to the panel surface must start not in the room air, but at the panel surface.

Specifically, therefore, the problem of panel rating is visualized as requiring the determination of a conductance rather than an overall heat transfer coefficient; the conductance of the panel from the water within the tubes to the panel surface. Thus if t_w and t_p are taken, respectively, as the average temperatures of water passing through the coil length associated with unit panel area and the average surface temperature of that unit area, and if Q_p is the output of unit area expressed in Btu per hour, then the panel conductance, C, will be defined by the equation,

$$C = Q_P / (t_W - t_P) \qquad (1)$$

Since the conductance does not include conditions exterior to the panel it follows that for fixed values of t_w and t_p the experimentally determined value of C for a particular tube size and tube spacing should have a fixed value independent of the location (as floor, wall, ceiling) of the test unit.

METHOD

The panel test station was set up inside the air conditioning research rooms at the University of California; these rooms, described in a previous paper,[2] are themselves located within a large laboratory structure. Air within the research rooms could be maintained at any desired temperature between 40 F and 140 F with ventilation at any selected rate between zero and fifteen air changes.

To permit exact duplication of procedure for panels using various sizes and spacings of tubing a standard size of 4 ft x 4 ft was selected for each test section. Fig. 1 is a rear view of a representative test panel prior to application of the plaster. Coils were located *above* the metal lath and each panel was contained within a 2 in. x 4 in. wooden frame with steel supporting cross-bars from which the coil and plaster were suspended. Back-plastering assured complete embedment of the coils. Plastering was done in the usual way, but under controlled conditions so that the uniformity of the work was much greater than that usually found in practice. Fig. 2 shows a front view of the test surface of a typical test panel.

The test station consisted of a 6 ft x 6 ft x 8 in. cork block, finished on the bottom and sides with plywood and with a 4 ft x 4 ft x 4 in. pocket in the bottom into which the various standard size test panels could be inserted. The cork block was supported on pipe legs at a distance of 8 ft from the floor, thus constituting a 36 sq ft element of hung ceiling with an inserted 16 sq ft heating panel. Heat losses from the sides of the test unit were thus retarded by 12 in.

of cork, whereas a 4 in. cork barrier retarded losses from the rear of the panel. Fig. 3 is a photograph of the test station taken, looking up, from the floor of the air conditioning research room. The white central section is the heated ceiling surface and the six inch vertical lip shown around the plywood-finished cork box is a barrier intended to prevent heated air from sliding out from under the test panel and rising to the room ceiling. Twenty-eight thermocouples are shown attached to the test surface as also is a 4 in. x 4 in. Gier and Dunkle heat meter.

Because of the fact that it was desired to run tests with widely different flow rates, a decision was reached to use a once-through flow circuit with water from

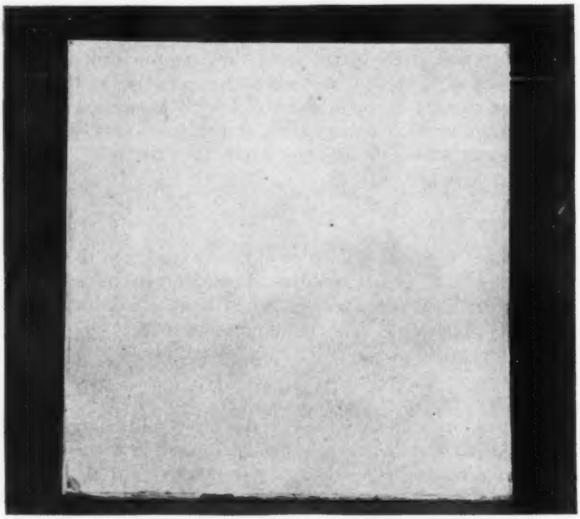


FIG. 2. FRONT VIEW OF TEST PANEL SURFACE

the test panel going through a weighing tank and then to discard in the building drain. Cold water from the city system flowed continuously through a standard residential gas-fired hot water heater, leaving at a temperature a few degrees lower than that desired at the panel and discharging from the heater to an open 100 gal tank which was located on top of the research rooms. Electric immersion heaters, under automatic control, raised the temperature in the tank to the desired value and maintained it at a fixed setting within 1/5 F deg; mechanical agitation was used to assure uniformity of temperature throughout the mixing tank.

From the mixing tank flow occurred by gravity down to the research room, through the test coil and into the weighing tank; a valve in the line at discharge from the test panel permitted control of the flow rate with exact adjusment by means of a needle valve located in a bypass around the main valve. Temperature of the water to and from the test coil was determined by oil-immersed

calibrated mercury thermometers (one-tenth degree least count) and checked with copper-constantan thermocouples. Thermocouples located at intervals in the cork block were used to check the temperature gradient through the side and rear insulation and these calculations were checked by means of direct heat meter readings from the exposed surfaces of the insulation.

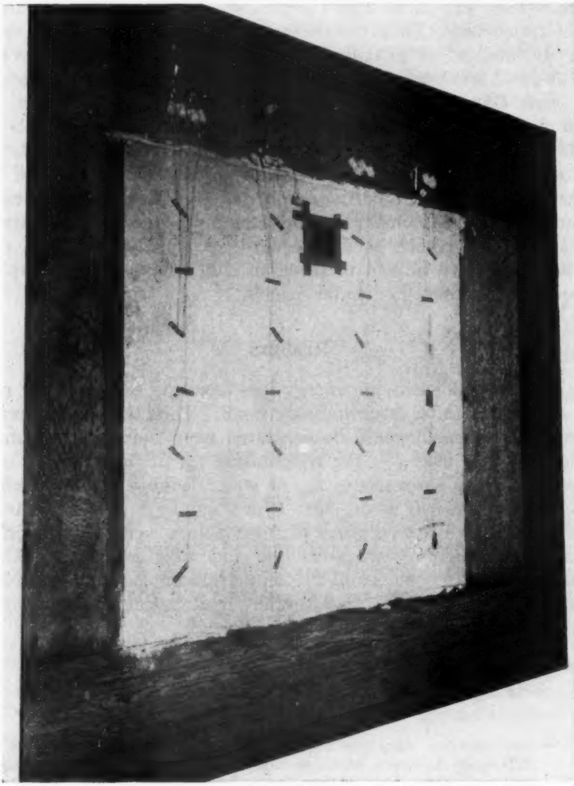


FIG. 3. BOTTOM VIEW OF TEST PANEL LOCATED IN CEILING

Nine test panels were constructed using ⅜, ½, and ¾ in. Type K copper tubing on 4, 6, and 9 in. centers. As shown in Figs. 1 and 2, all panels were of the sinuous coil type and supply and return connections were arranged to protrude through prepared openings (sealed with ground cork) in the block of insulation. Because of the not inconsequential weight of the test sections the cork block was arranged so that one section could be removed (1 ft x 4 ft piece in the foreground of Fig. 3) to permit hoisting the test specimen to the required

elevation and then sliding it into place on rails built into the insulation. Prior to testing, each unit panel was sealed into the insulation by filling the small cracks (approximately ⅛ in.) that had been left to provide clearance between the panel and the cork block.

Data were recorded for three identical tests on each of the nine panels and in no case was a record made until a minimum of 24 hr after test conditions had been impressed on the panel and a separate minimum of 6 hr after the last evidence of transience. To avoid the possibility of error through inadvertent change in the method of installing or sealing the panels, the three tests on each type of panel were not run in progression, but three complete sequences of nine tests each (covering all panels in each sequence) were made. In addition to the nine plaster panels, units were also constructed with coils embedded in concrete and coils standing in free air back of a plywood surface; one unit consisted of a coil in loose sand (back of plywood) and others used tube sizes greater than ¾ in., but neither the number of test panels of a given type nor the number of check tests on these nonplaster units were great enough to warrant inclusion of the results in this paper. Specifically, therefore, the results given in this paper are limited to three sizes of tubing on three spacings for sinuous coil, copper tube, hot water panels.

RESULTS

For equilibrium conditions the energy loss from the sides and the rear of the test panel was found to be insignificantly small. Thus the desired experimental result, the conductance C, could be evaluated from four experimentally determined values. These are: (1) the temperature t_{wi} of the water at entrance to the test coil; (2) the temperature t_{wo} of water leaving the test coil; (3) the flow rate, W, expressed in pounds per hour; and (4) the surface temperature of the test panel, t_p. The first three of these values were determined by direct observation (from mercury thermometers and weighing tank), whereas the fourth was taken as the average of the 28 thermocouple readings. Noting that the mean water temperature, t_w, is the arithmetical average of t_{wi} and t_{wo} and substituting into Equation 1,

$$C = \frac{2W\,(t_{wi} - t_{wo})}{A_p\,[\,(t_{wi} + t_{wo}) - 2t_p)\,]} \qquad (2)$$

where

C = conductance, Btu per (square foot of panel) (hour) (Fahrenheit degree difference between average temperature of water in coil and average surface temperature of panel)

W = water flowing through coil, pound per hour

t_{wi} = temperature of water entering coil, Fahrenheit degree

t_{wo} = temperature of water leaving coil, Fahrenheit degree

t_p = temperature of surface of test panel, Fahrenheit degree

A_p = area test panel, square feet

Table 1 gives the resulting values of C for the nine test panels.

Taking ½ in. tube on 6 in. centers as an arbitrary standard, the value of C for this test panel was found to be 1.55 Btu per (hr) (sq ft) (F deg). Tests

of the same spacing with tube sizes of ⅜ in. and ¾ in. gave *C* values less and greater, respectively, by 2½ percent. The absolute values of the conductances for panels with tube spacings other than 6 in. were not the same, but the percentage change in conductance due to increasing the size from ½ to ¾ in., or in decreasing it from ½ to ⅜ in. was, for all three spacings, found to be practically equal to 2½ percent. Thus it appears from these tests that tube size, within the range from ⅜ to ¾ in., has no appreciable effect on the rate of energy dissipation from panels using sinuous coils with spacings of 4, 6, or 9 in. On the basis of Btu transferred per pound of copper, the smaller size tubing evidently has a substantial advantage. In practice, however, a selection of tube size must be tempered by other considerations such as pressure drop, maximum permissible temperature difference between entering and leaving water, uniformity of panel surface temperature (which may have a definite influence on the life of the plaster) and other considerations.

TABLE 1—RESULTS OF THERMAL CONDUCTANCE[a] TESTS OF 9 SINUOUS COIL, TYPE K COPPER TUBE PLASTER PANELS

TUBE DIAM. IN. O.D.	SPACING OF TUBES (CENTER TO CENTER) IN.		
	4	6	9
⅜	2.32	1.50	1.07
½	2.39	1.55	1.11
¾	2.48	1.60	1.15

[a]Btu per (hr) (sq ft of panel surface) (F deg temperature difference between mean temperature of water in the coils and the average surface temperature of the panel)

For fixed tube size and variable spacing it was again found that a single correlation applied to all three sizes. Taking 6 in. spacing as an arbitrary standard, the value of *C* was found to be reduced to 72 percent when the spacing was increased to 9 in. and raised to 154 percent when the spacing was reduced to 4 in. Expressed differently, the transfer rate per lineal foot of tubing was 100 percent when centered at 6 in., 2½ percent less for 4 in. centers, and 7 percent greater for 9 in. centers. As a very rough approximation (accurate to 90 percent) the transfer rate, over the range considered, was essentially independent of both spacing (as such) and tube size and retained a fixed value of approximately 0.77 Btu per hr per *lineal foot* of tube per 1 F deg temperature difference between water and panel surface.

CONCLUSIONS

When a panel heating design problem has been attacked by the heat-balance method, the designer obtains as part of his answer the required area of panel which, for design conditions, must operate at the design maximum surface temperature. An additional piece of information available from the heat balance analysis is the required heat output, expressed in Btu per (square foot of panel) (hour). As a first, and usually conservative, approximation it is suggested that the known panel rating be divided by 0.77 to obtain a number which can be taken as equal to the product of the water-to-surface temperature difference and the required lineal feet of tubing per square foot of panel. Now either the

water temperature can be arbitrarily fixed and the tube spacing calculated, or vice versa.

REFERENCES

1. Panel Heating and Cooling Analysis, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 47, 1941, No. 1182, p. 285.)

2. A.S.H.V.E. RESEARCH REPORT No. 1192—Panel Heating and Cooling Performance Studies, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, p. 35.)

3. Trend Curves for Estimating Performance of Panel Heating Systems, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 48, 1942, No. 1217, p. 425.)

4. A.S.H.V.E. RESEARCH REPORT No. 1252—Optimum Surface Distribution in Panel Heating and Cooling Systems, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 50, 1944, p. 231.)

5. A.S.H.V.E. RESEARCH REPORT No. 1274—Radiation Corrections for Basic Constants Used in Design of All Types of Heating Systems, by B. F. Raber and F. W. Hutchinson. (A.S.H.V.E. TRANSACTIONS, Vol. 51, 1945, p. 213.)

1323

No. 1323

EFFECT OF FLOOR SLAB ON BUILDING STRUCTURE TEMPERATURES AND HEAT FLOW

By CARL F. KAYAN,* NEW YORK, N. Y.

COMPLEX heat flow paths are to be found in building structures in a wide variety of forms and arrangement. Under such circumstances it is difficult, and generally impractical, to calculate steady-state heat flow and surface temperature distributions by orthodox, mathematical or graphical techniques, particularly when surface conductances are involved on the high and low temperature sides.

Especially difficult is the problem when the heat flow lines, by virtue of the complex heat path, are distorted. Herein the method of electrical analogy, applied to a geometrically similar model, offers the possibility of prediction of thermal values of heat flow and temperature distributions for the actual structure. It is to be noted at the very outset, however, that numerical values of surface conductance and wall conductivity must be assumed for the purposes of this procedure, similarly as in any other calculation technique; these assumptions are reflected in the final result.

One particular structural arrangement of interest is that representing an outside uniformly thick concrete wall with its attached floor slab, as shown in Fig. 1. For the case chosen, the outside surface is in contact with outdoor air at 0 F, while the inner surfaces are exposed to indoor air at 70 F. Under these circumstances, presuming steady-state conditions, heat will flow through the wall—at some distance away from the horizontal floor slab—in undistorted fashion: the isotherms will be parallel to the wall surfaces. This is in accord with results of usual calculation technique. However, as the locale of the floor slab is approached, the conditions become more complicated, due to the heat flow from the slab in contact with the indoor air. Relative to the wall and in terms of heat transfer structure nomenclature, the slab in effect is an extended surface fin and thus represents a two-dimensional problem.

It is obvious that the surface temperatures of the vertical wall where one-dimensional heat flow takes place will be different from those of the slab some

*Associate Professor of Mechanical Engineering, School of Engineering, Columbia University. Member of A.S.H.V.E.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Coronado, Calif., June 1947.

distance in, where no heat flow takes place by virtue of the slab being at the air temperature. In between these two extreme positions, the structure surface temperatures will vary, and such temperatures are difficult of prediction. With growing interest in radiant panel heating, involving calculation of mean radiant surface temperatures, knowledge of the structural surface temperature is highly desirable in order to permit estimation of values. Given conditions of quiescent air, with stratification such that the air above the floor is of lower temperature than that of the air below, the situation becomes still more complicated.

It is to be noted that the heat transfer surface conductance varies primarily with the character of the surface temperature levels concerned, and with the

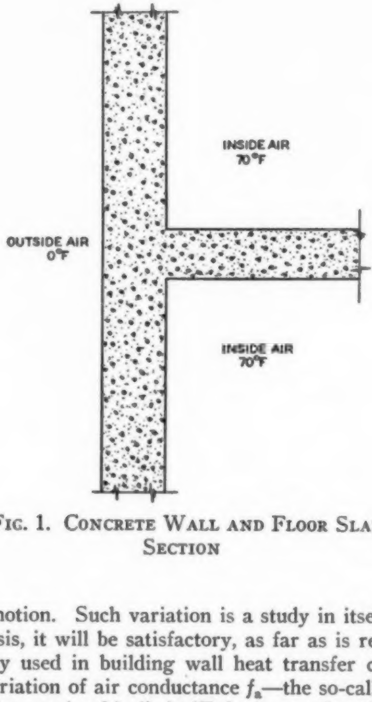


FIG. 1. CONCRETE WALL AND FLOOR SLAB SECTION

velocity of the air motion. Such variation is a study in itself. For the purpose of the present analysis, it will be satisfactory, as far as is reasonable, to employ values conventionally used in building wall heat transfer calculations. Fig. 2 shows the typical variation of air conductance f_a—the so-called heat transfer air film coefficient, in Btu per (sq ft) (hr) (F deg), as a function of wind velocity, from Rowley et al, reported in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1947 and elsewhere,[1] and is particularly intended for outside air conditions. The corresponding value of R_a—thermal resistance per square foot, F deg (sq ft) (hr) per Btu, has been added in Fig 2.

Conventionally, the conductance f_i for the inside air is taken for still air and is customarily cited at a uniform value of 1.65 Btu per (sq ft) (hr) (F deg).

[1]Exponent numerals refer to Bibliography.

Herein the values are open to question, particularly when the varying radiation effect of confronting surfaces and the relatively stagnant conditions of corner air are to be given consideration, and are certainly to be assumed as different from the customary value of 1.65.

The influence of radiation on the surface film coefficient under different conditions of temperature for confronting surfaces has been described in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1947.[3] Heat flow values are given for a vertical surface at 80 F, with ambient still air at 70 F, and effective

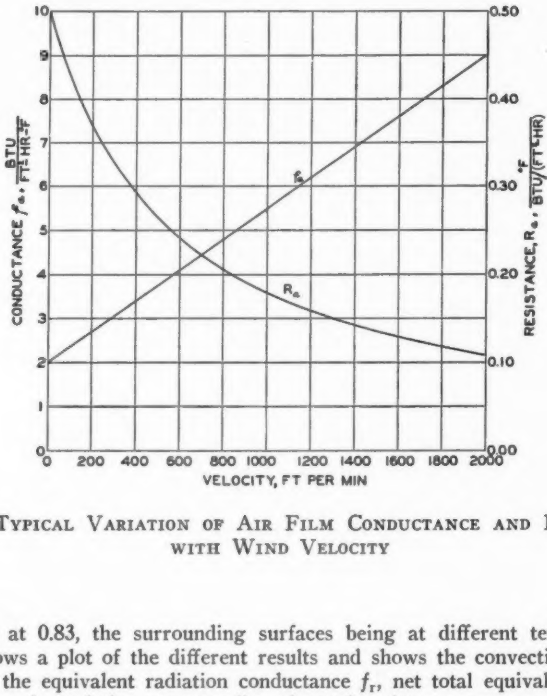


FIG. 2. TYPICAL VARIATION OF AIR FILM CONDUCTANCE AND RESISTANCE WITH WIND VELOCITY

emissivity at 0.83, the surrounding surfaces being at different temperatures. Fig. 3 shows a plot of the different results and shows the convection conductances f_c, the equivalent radiation conductance f_r, net total equivalent surface conductance f_n, and the corresponding thermal resistance per square foot R_n. With differing wall temperatures prevailing in a room, the importance of the actual value of f_n is apparent. It should, of course, be noted that the radiation conductance f_r could have a negative value.

In view of the foregoing circumstances of stagnant corner air, it is reasonable to assume that the effective surface conductance will be considerably reduced as the corner is approached. Fig 4. shows an assumed variation of f_1 as a function of distance from the corner, along with its corresponding value of unit thermal resistance R_1 (further experimental work to establish this variation is desirable and is separately under study).

EXPERIMENTAL PROGRAM

The purpose of this paper is to investigate temperature distribution and heat flow for a single case of a concrete wall with its attached floor slab shown in Fig. 1. Different conditions are presumed to exist for the outside conductance f_o, for which the typical variation is given in Fig. 2. The inside conductance f_i is assumed to vary in accordance with the values as given in Fig. 4, rising to a

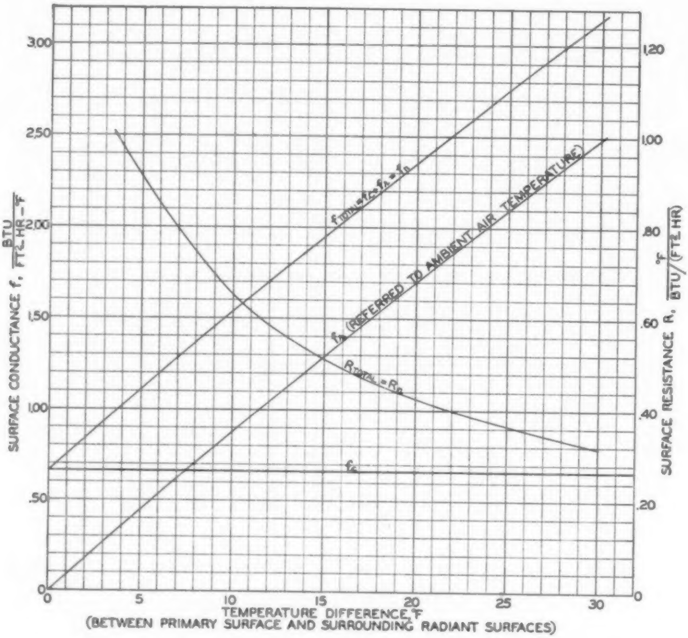


FIG. 3. INFLUENCE OF RADIATION ON SURFACE FILM COEFFICIENT

constant value of 1.65 for quiescent air at a distance of 8 in. from the inner corners, the same values being taken for both the horizontal and the vertical surfaces. The conductivity of concrete is assumed as $k = 9.00$ Btu per (sq ft) (hr) (F deg per in.); the wall is 8 in. thick, and the attached floor slab 6 in. Inside air is taken at 70 F, outside air at 0 F, with three separate values of f_o of 2.00, 3.00, and 6.00. An additional case is analyzed in which the floor air is presumed to be lower in temperature than the ceiling air by 10 percent, with the average of the two at 70 F, thus accounting for some air stratification. This case has been studied with f_o at 6.00.

Isotherms are plotted to illustrate the distortion of temperature lines by the presence of the attached fin-like slab, as compared with those resulting in the

wall some distance away from the slab, where they are regularly spaced and parallel to the wall surfaces. Fig. 5 shows the temperature distribution fraction as a function of thermal resistance R for two cases of $f_0 = 2.00$ and 6.00, from which the corresponding wall surface temperatures for one-dimensional flow may be inferred; the resistance values are in proper relationship to one another although their values are arbitrary.

The investigation has been carried on through the use of the resistance-concept Analogger previously described by the author.[3] Briefly, this embodies a geometrical type of analogue for steady-state conditions based on the resistance concept which recognizes the general similarity between heat flow and electrical flow. It is to be noted that the use of electrical analogies for simple cases is not new, particularly as involving isothermal surfaces, and there are references in the literature (more particularly in the European) starting with the work of Langmuir[4] here in 1913. The heat flow model of the Analogger† has the same geometrical configuration as the original structure cross-section, and uses electrically conductive flat sheet along which electrical flow takes place.

APPLICATION OF THE ELECTRICAL ANALOGGER

The geometrical electrical analogy facilitates the experimental procedures and permits not only ready prediction of temperatures throughout a heat flow path, but also quantitative prediction of heat flow in accordance with assumed conditions. Basically it recognizes the application of the so-called Ohm's law for electrical circuits to equivalent thermal circuits.

The relationships are easily stated:

$$i = \frac{\Delta e}{r};\ e = ir \qquad (1)$$

where

i = electrical current, amperes

Δe = potential difference, volts

r = electrical resistance of the flow circuit, ohms

Similarly, the rate of heat flow depends on temperature difference and the equivalent resistance to heat flow; this principle is identified as the resistance concept of heat transfer. Thus:

$$H = \frac{\Delta t}{R};\ \Delta t = HR \qquad (2)$$

where

H = heat flow rate per unit of area, Btu (sq ft) (hr)

Δt = overall temperature difference, F

R = thermal resistance per unit of area, deg F (sq ft) (hr) per Btu

In addition, to cover the case of overall conditions for area A:

$$q = HA = \frac{\Delta t A}{R} \qquad (3)$$

†Name coined to represent the equipment for logging the isopotentials and resistances.

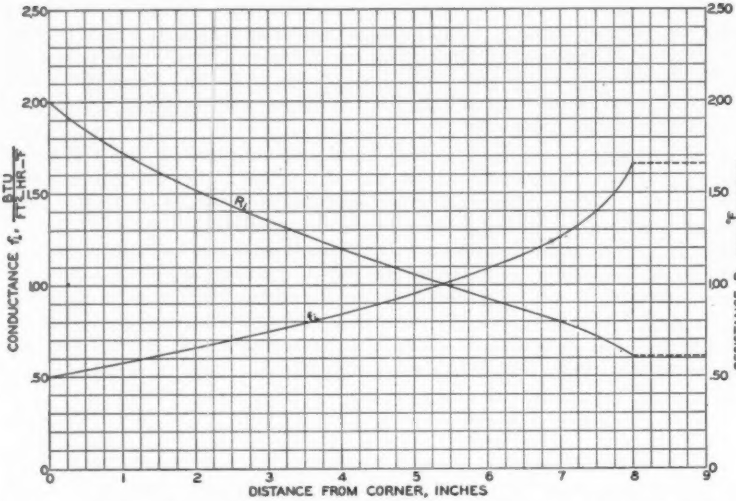


FIG. 4. ASSUMED VARIATION OF FILM CONDUCTANCE WITH DISTANCE FROM CORNER FOR CONSTRUCTION SHOWN IN FIG. 1

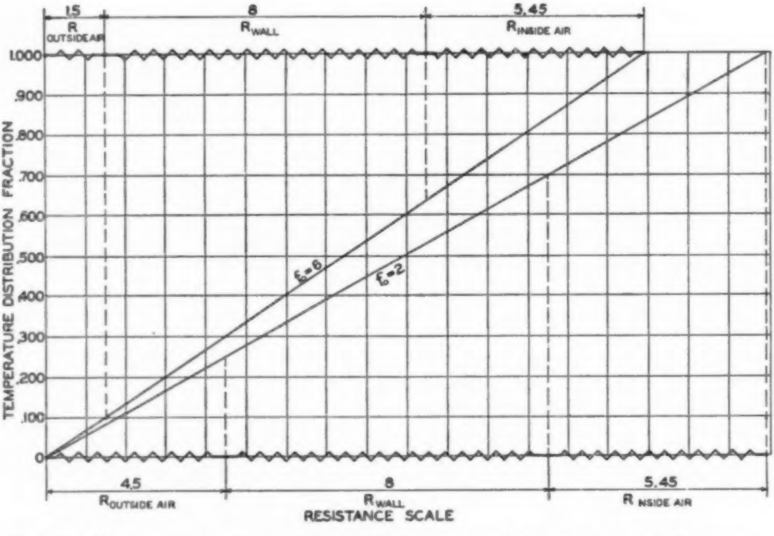


FIG. 5. VARIATION OF TEMPERATURE DISTRIBUTION FRACTION WITH THERMAL RESISTANCE FOR $f_o = 2$ AND $f_o = 6$

where

q = gross heat flow, Btu per hr
A = heat flow reference area, sq ft

For a heat flow path between two temperature levels, equivalent temperature conditions throughout may be determined from the simulating electrical flow path, if the component electrical resistances have the same relationship between themselves as have the thermal resistances. Accordingly, for the electrical model

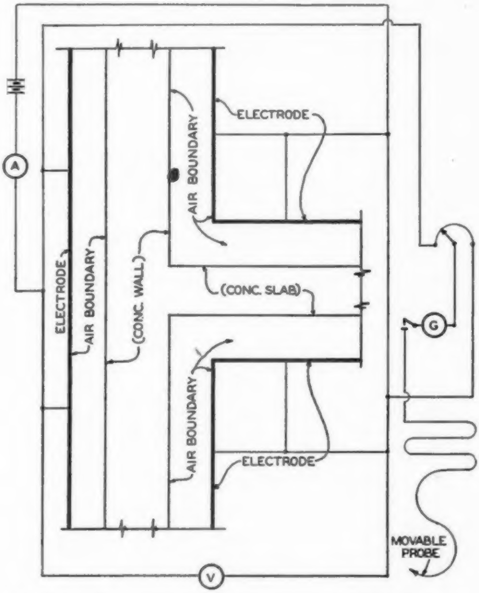


FIG. 6. ELECTRICAL CIRCUIT OF ANALOGGER IN TEST OF CONCRETE WALL AND SLAB SECTION

representing a wall transferring heat, electrical potentials throughout will be representative of the temperature conditions.

Thus the problem consists in setting up an electrical model in which all the thermal resistances of the heat transfer arrangement are represented by appropriate electrical resistances. This is accomplished in the Analogger through the use of electrically conductive flat sheet with intimately contacting electrodes mounted thereon to represent the conditions of the fluid boundaries of the thermal structure. In using uniform sheet, linear distances on the sheet are commensurate with resistances. The pattern of the heat transfer wall is thus laid out in geometrical equivalence thereon. Offsetting the electrode on the sheet by a given distance from the wall profile is equivalent to introducing an

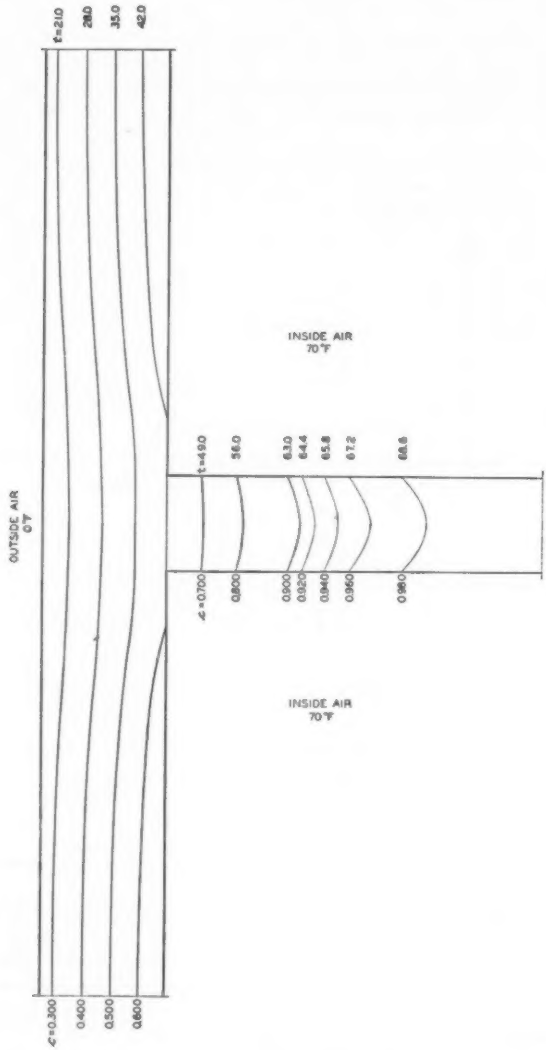


FIG. 7. ISOPOTENTIAL PATTERN FOR SERIES I ($f_o = 2$)

additional resistance to heat transfer between the boundary fluid at its prevailing temperature and the wall itself.

The general fluid boundary resistance R may be defined through the fluid conductance f:

$$R = \frac{1}{f} \qquad (4)$$

where

R = equivalent boundary resistance per sq ft, deg F (sq ft) (hr) per Btu

f = surface conductance (*film coefficient*), Btu per (sq ft) (hr) (F deg)

The thermal resistance R_w of wall material of uniform composition is

$$R_w = \frac{x_w}{k_w} \qquad (5)$$

where

x_w = thickness of wall material, inches

k_w = thermal conductivity of wall material, Btu per (sq ft) (hr) (F deg per in.)

For a given value of f, such as f_a for an air boundary, having its corresponding value of R_a, there is some equivalent thickness x_e of wall material which will give the same resistance R_e to heat transfer as the fluid boundary:

$$R_a = \frac{1}{f_a} = R_e = \frac{x_e}{k_w} \qquad (6)$$

From this it follows that

$$x_e = \frac{k_w}{f_a} \qquad (7)$$

Thus the equivalent electrical resistance to permit simulation of thermal conditions on the Analogger would be proportional to x_e and x_w to represent the fluid and the wall, if just a simple wall and fluid were in question. When using flat, electrically conductive and uniform sheet for one- and two-dimensional Analogger studies, these resistances could be represented by proportional distances or lengths l_e and l_w, thus permitting a model of the heat flow path thermally in scale.

EXPERIMENTAL PROCEDURE

In the present problem, an 8 in. wall section has been analyzed for a 60 in. height, the 6 in. slab of 23 in. length being joined to it at the mid section and of uniform homogeneous material of the same conductivity, $k_w = 9.00$, as the wall itself. The Analogger model for convenience was made full scale, with distances on the sheet equal to the structure dimensions. Its electrical circuit is shown in Fig. 6. Computation of the resultant electrical resistance of the model between the electrodes for the different arrangements is possible through the determination of the overall voltage and the circuit current as supplied by the electrical source. By means of a parallel slide wire arrangement, deflecting galvanometer, and movable probe, potential distribution over the electrical sheet

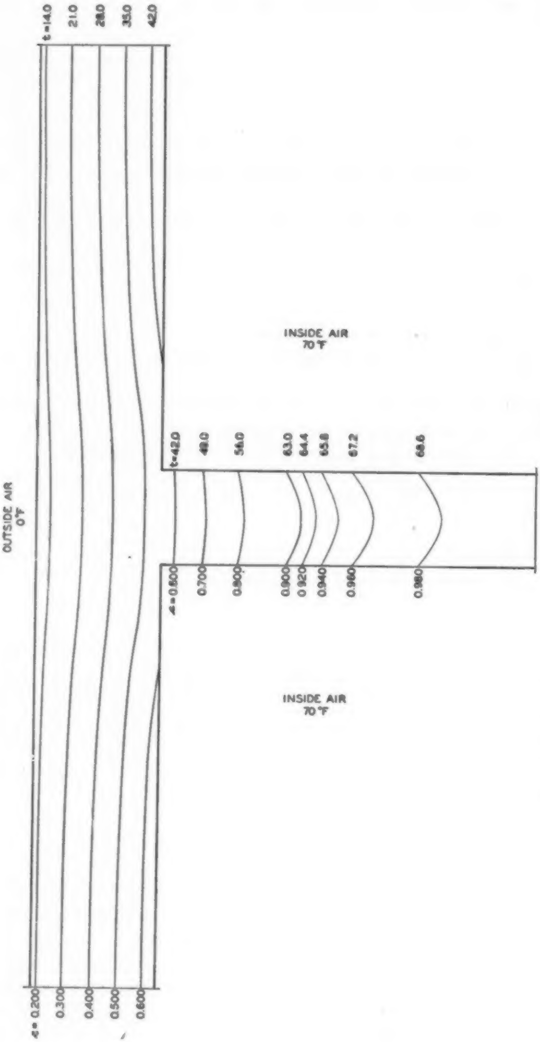


FIG. 8. ISOPOTENTIAL PATTERN FOR SERIES II ($f_o = 3$)

was determined. In this connection, use was made of a potential difference fraction defined as follows:

$$c = \frac{\Delta e_p}{\Delta e} \qquad (8)$$

where

Δe_p = electrical potential difference from zero base to a given point p

Δe = overall electrical potential difference between electrodes

Thus, c, the fraction of the overall potential difference at any point, could be determined and the isopotential lines developed thereby. Equivalent temperatures followed from values of c and the overall temperature difference:

$$t_p = t_o + c\Delta t_{o\text{-}i} \qquad (9)$$

where

t_p = temperature at a given point p, Fahrenheit

t_o = temperature of outside air, Fahrenheit

$\Delta t_{o\text{-}i}$ = temperature difference between inside and outside air, F deg

Thus for 70 F temperature difference with outside air at 0 F, for $c = 0.400$ at a given point, p, $t_p = 0 + 0.400 \times 70 = 28.0$ F.

As previously noted, the distance of offset for the outside-air electrode from the wall profile depends on the conductance f_o. For values of f_o = 2.00, 3.00, and 6.00 to be studied, with the Analogger model made to scale, the corresponding values of l_e in accordance with Equation 7 were 4.5, 3.0, and 1.5 in., it being recalled that $k_w = 9.00$. For the inside air, f_i was taken as 1.65 for those sections more than 8 in. away from the corner, thus for the model, l_e = 5.45 in. on both wall and slab surfaces. For the inner 8 in. to the corner, the equivalent resistance in accordance with the assumed curve of Fig. 4 was obtained by appropriate cutting of the boundary sheet material to produce resistance effects in accordance with the curve. Throughout, the sheet material representing the air boundaries was slit to prevent lateral electrical flow, thus to more adequately simulate surface conductance effects.

RESULTS OF INVESTIGATION

The resultant isopotential patterns are shown in Figs. 7, 8, and 9 for Series I ($f_o = 2$), Series II ($f_o = 3$) and Series III ($f_o = 6$) respectively. Here the distortions due to the slab heat flow for the isopotential lines and the corresponding isothermal lines may readily be observed. The variation of the temperature across the vertical sections of the slab is distinctly evident. From this it may be parenthetically noted that the conventional assumption of uniform temperature at any section so often made for the mathematical solution of fin problems is here not sustained. (The usual mathematical assumption of uniform surface conductance along the *fin* was not allowed here.)

The effect of different outside conductances f_o and their equivalent resistances is to be noted in comparison of these three figures. At the ends of the wall sections remote from the slab, the isotherms become straight, and position in accordance with conventional one-dimensional heat flow, such as indicated in

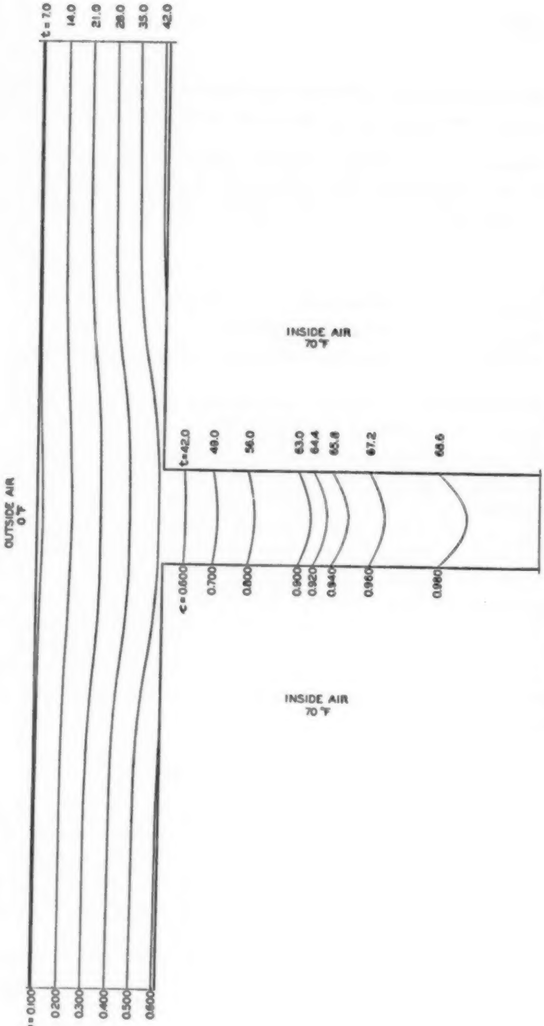


FIG. 9. ISOPOTENTIAL PATTERN FOR SERIES III ($f_o = 6$)

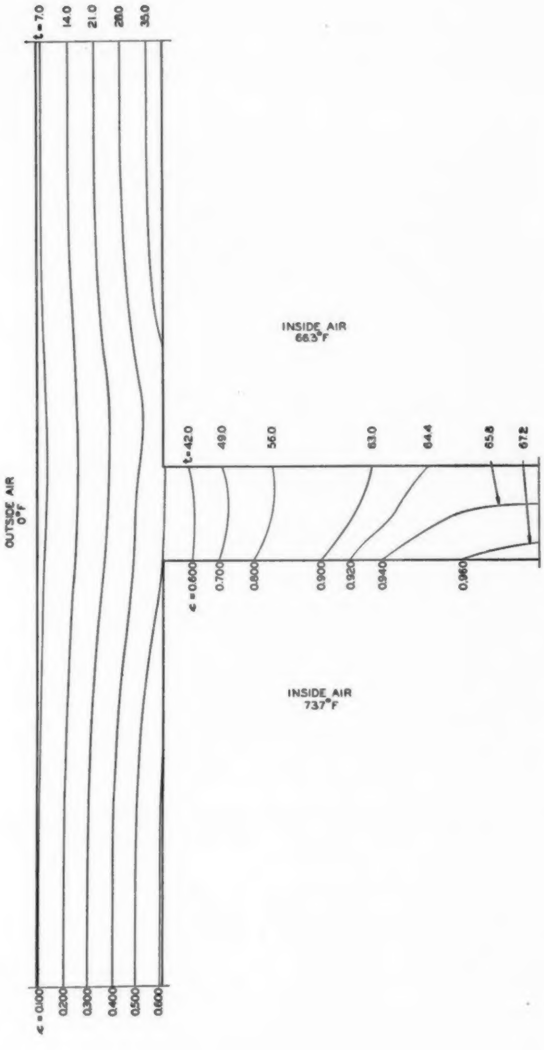


FIG. 10. ISOPOTENTIAL PATTERN FOR SERIES IV ($f_o = 6$) WITH AIR TEMPERATURE DIFFERENCE ABOVE AND BELOW SLAB

Fig. 5. The patterns for Figs. 7, 8, and 9 are symmetrical about the horizontal axis through the slab.

Of considerable interest are the results of Series IV, given in Fig. 10. As previously noted, the floor air temperature is taken as lower than that of the ceiling, to account for room air stratification. Specifically, the air in the ceiling section is uniformly taken at 73.7 F, and that in the floor section at 66.3 F (90

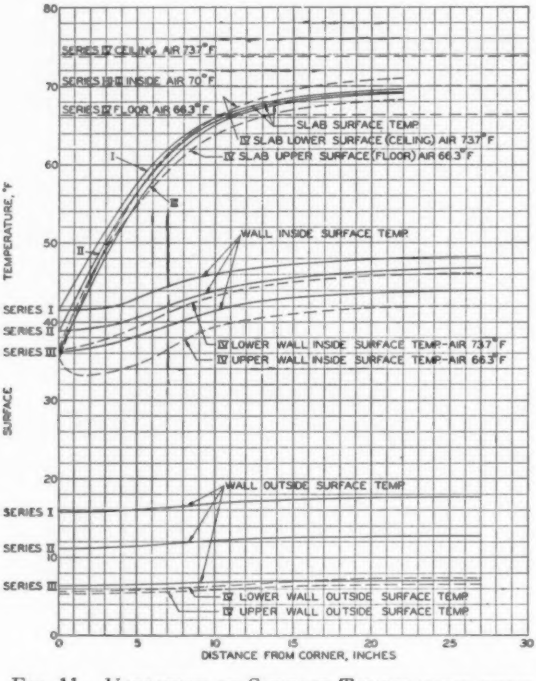


FIG. 11. VARIATION OF SURFACE TEMPERATURE WITH DISTANCE FROM CORNER (FOR CONSTRUCTION SHOWN IN FIG. 1)

percent of 73.7); the average is 70 F. The values of c are referred to 73.7 as the basic value. Again here the isotherms flatten out to their normal positions at each end of the wall. The floor slab to some extent transmits heat from the underside to the top. The patterns here are non-symmetrical, and the distortions are hardly predictable with facility by any other conventional type of analysis.

The surface temperature results for the different series are given in Fig. 11 and emphasize the effect of the slab heat flow on the surface temperature distributions. Despite the adjustment in the value of f_i to account for corner stagnation,

as given in Fig. 4, the slab surface temperatures present a general pattern customarily expected for fins.

From the results of the overall resistances in the different cases, predictions of ultimate heat flow through the combined slab-wall thermal path are possible. Following the methods previously described by the author in another paper,[5] the resistance results are correlated by plotting the ratio of electrical resistances

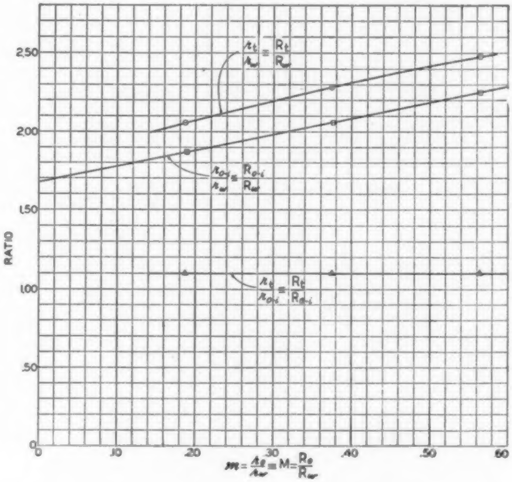


FIG. 12. VARIATION OF RATIOS $\dfrac{R_t}{R_w}$ $\dfrac{R_{o-i}}{R_w}$ AND $\dfrac{R_t}{R_{o-i}}$ WITH CHANGE IN RATIO $\dfrac{R_w}{R_o}$

against an operating characteristic m. Specifically, ratio r_t/r_w is plotted against $m = r_o/r_w$. Here, for Fig. 12:

r_t = measured electrical resistance between the electrodes for the full scale model under different conditions

r_w = electrical resistance of the model strip representing the plain wall section 8 in. thick

r_o = electrical resistance of the model strip l_e representing the outside air, variable with f_o

Another line is shown representing the variation of r_{o-i}/r_w with m, where r_{o-i} = electrical resistance of a uniform model strip for one-dimensional flow, comprising the series resistances equivalent to the outside air, wall and inside air for the full length of the 60 in. wall without a slab. A further line shows

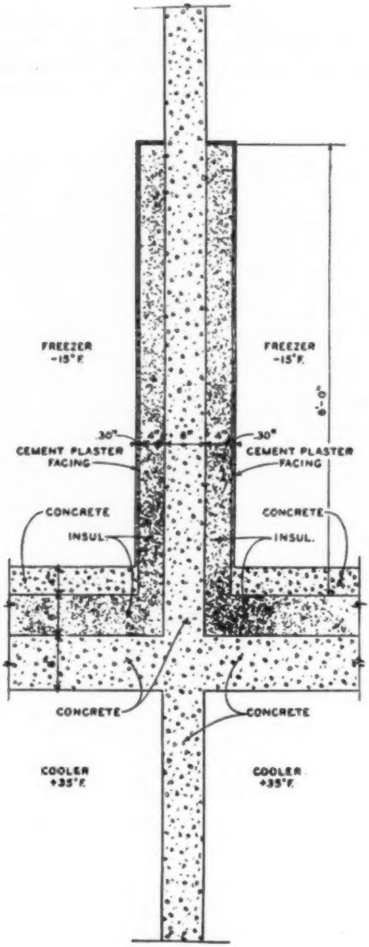


FIG. 13. COMPLEX COLD STORAGE WALL

the ratio of r_t/r_{o-i} vs. m, and in effect this may be directly translated into the equivalent thermal resistance ratio R_t/R_{o-i}

$$R_{o\text{-}i} = \text{series total for } R_o + R_w + R_i$$

for the actual slab-wall combination as compared with the same length plain wall having $f_i = 1.65$, and f_o as selected,

where

R_t = equivalent thermal resistance of wall slab assembly referred to average unit area for the combination

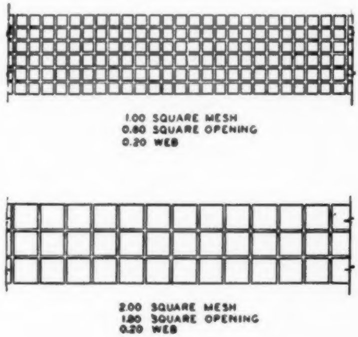


FIG. 14. PERFORATION OF ELECTRICAL CONDUCTING SHEET TO SIMULATE INCREASED RESISTANCE OF INSULATION

It will be apparent then that the thermal counterparts of the previous electrical values will be related to m as given by the equation:

$$m = M = R_o/R_w \quad \text{(10)}$$

where

M = operating characteristic for thermal transfer corresponding to m for the electrical system

$R_o = 1/f_o$

$R_w = \dfrac{x_w}{k_w}$

In effect then, the transition is made between the electrical results and the equivalent thermal resistance ratios for the wall arrangements in question. Thus heat flow values, in terms of unit areas of wall surface, may be compared with those resulting for a simple wall in one-dimensional heat transfer. Furthermore, since dimensionless values are employed, Fig. 12 offers the possibility of applying these results to other combinations of physical properties for the same general shape configuration.

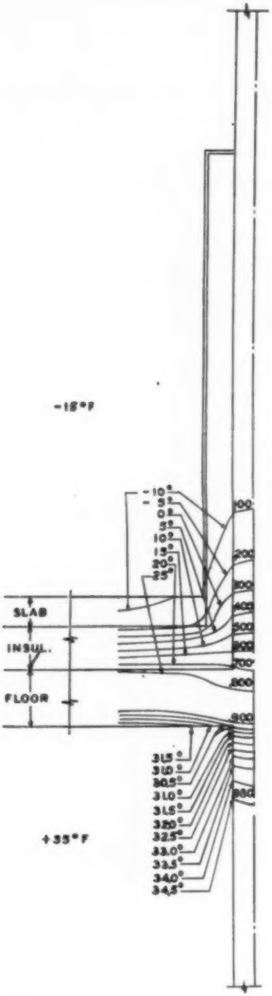


Fig. 15. Isotherms for Insulation to Wall Material Resistance Ratio of 4.6

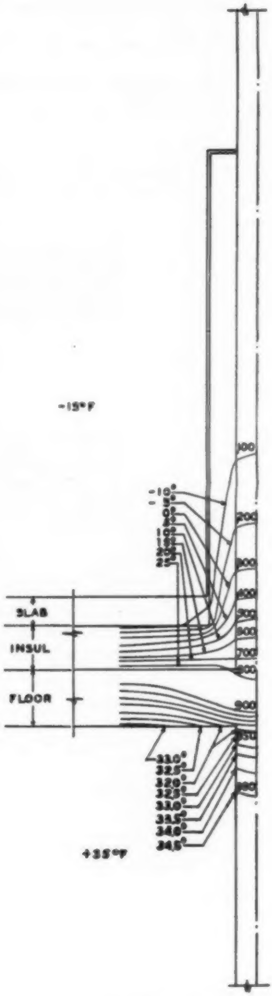


Fig. 16. Isotherms for Insulation to Wall Material Resistance Ratio of 9.2

COMPLEX CASE WITH MULTIPLE WALL MATERIALS

It is of interest to briefly mention another case studied in the Analogger Laboratory, involving heat flow analysis for a wall arrangement comprised of structural and insulating materials, and previously described by the author.[3] This concerns a cold storage application, and though of different temperature range from the preceding case, bears a relationship to it. Fig. 13 shows the structural arrangement, with *freezer* and *cooler* cold storage spaces presenting the different temperature levels. In detail this differs from the present case in that the complex heat flow path involves wall material of differing conductivity.

Since the structural arrangement was symmetrical about the vertical axis, only one side required study. The insulation, of lower conductivity than the concrete wall, required that in the flat sheet model the characteristics of the sheet section for the insulation be different from that for the wall. The problem that this offered was ultimately solved by perforating the sheet for insulation simulation in a square mesh pattern, such as shown in Fig. 14. Here the sheet resistance was distinctly increased for the meshed section vs. the unmeshed, and could be altered further by going to progressively larger meshes, as indicated in the figure.

The particular objective in this case was to determine the surface temperatures on the underside so that the possibility of condensation and frost accumulation on the surfaces could be studied. Figs. 15 and 16 depict the isotherms and the surface temperatures for the cases of one-inch and two-inch mesh, equivalent to resistance ratios vs. wall material of 4.6 and 9.2, respectively.

In conclusion, it must again be emphasized that the results obtained depend directly on the assumptions for the physical factors of wall conductivity, and inside and outside boundary conductances.

ACKNOWLEDGMENT

Acknowledgment is hereby made to different associates of the author who have cooperated on various aspects of the investigation: to Barbara Sherman Kayan for development of the isopotential patterns, and to R. G. Vanderweil, Chase Brass and Copper Co., Waterbury, Conn., who among others suggested the study of the problem.

This paper reports the results of investigations supported in part by a grant-in-aid to Columbia University from The Refrigeration Research Foundation, Inc.

BIBLIOGRAPHY

1. HEATING, VENTILATING, AIR CONDITIONING GUIDE 1947, p. 122; also A.S.H.V.E. RESEARCH REPORT No. 869—Surface Conductances as Affected by Air Velocity, Temperature, and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429.)

2. HEATING, VENTILATING, AIR CONDITIONING GUIDE 1947, p. 121.

3. Temperature Distribution in Complex Wall Structures by Geometrical Electrical Analogue, by C. F. Kayan. (*Refrigerating Engineering,* February 1945, Vol. 49, pp. 113-117, 139.)

4. Flow of Heat Through Furnace Walls, by I. Langmuir, E. Q. Adams and F. S. Meikle. (*Transactions, American Electrochemical Society,* Vol. 24, 1913, pp. 53-84.

5. Temperature Patterns and Heat Transfer for a Wall Containing a Submerged Metal Member, by C. F. Kayan. (*Refrigerating Engineering,* June 1946, Vol. 51, No. 6, p. 553.)

DISCUSSION

JOHN EVERETTS, JR., San Francisco, Calif. (WRITTEN): I would like to compliment the author on the presentation and the analysis of a subject about which very little work has been done. This is, of course, a preliminary study of a subject which is important to anyone in the heating, air conditioning and refrigeration field.

Recently, I have been making a similar study from the standpoint of the formation of condensation in a problem much similar to that shown in the author's paper in Fig. 13. For this reason I would like to know if the author has been able to determine *when* condensation will form and *to what extent.* In my preliminary studies it was found that this problem is extremely complex because, as soon as condensation begins to form, the surface temperature approaches the wet bulb temperature instead of the dry bulb temperature of the air adjacent to that particular surface. Additional condensation continues to form because of this combination of cooling from the slab and cooling as indicated by the wet bulb temperature.

Since this problem of condensation is of definite importance when it occurs, I would like comments from the author on work of this type which he has done or anticipates doing.

C. B. BRADLEY, Manville, N. J. (WRITTEN): The author has illustrated the usefulness of his ingenious *Analogger* in solving practical problems. The three conditions discussed are all of practical interest. The last case, involving insulated walls and floors, serves to illustrate the effectiveness of the Analogger in handling problems of considerable complexity.

The curves of Fig. 4 in the preprint appear not to be drawn so as to be consistent with the text. The text states:

The inside conductance f_1 is assumed to vary in accordance with the values as given in Fig. 4, rising to a constant value of 1.65 for quiescent air at a distance of 8 in. from the inner corners . . .

In Fig. 4, however, f_1 has a value of about 1.45 at the 8 in. distance and does not appear to be approaching a constant value at any distance.

AUTHOR'S CLOSURE: I would first call attention to the following corrections that are necessary in the preprint of the paper:

Equation 1 should read:

$$i = \frac{\Delta e}{r};\ \Delta e = ir$$

In designation of units for resistance R in Figs. 2, 3 and 4, the symbol after "Btu" (in denominator) should be division sign "/" rather than dash.

Mr. Bradley is quite correct in pointing out the discrepancy in Fig. 4, where the values of the arbitrarily chosen curve for f_i at 7 in. from the corner should be 1.27; at 7.5, 1.40, and at 8, 1.65. The corresponding values for the resistance R should be 0.79, 0.71, and 0.61. Values outward from 8 in. on are constant and are taken as those for 8 in.

Mr. Everetts has drawn attention to a very significant problem occurring when cold surfaces are in contact with moisture-laden air. Up to the time when the surfaces in question are still dry, the air conductance is of low order and of value depending on the prevailing conditions of air motion and surrounding radiation. When condensation begins to form on the surfaces, however, the surface conductance changes and presumably goes up considerably, with resultant decrease in surface resistance. Under these circumstances, the nature of the heat flow in the area of condensation may then change, depending on the relative resistances of the thermal circuits, and accompanied by probable change in the surface temperatures. On the basis of Mr. Everetts' suggestion, the author plans to investigate these effects further.

No. 1324

TRIETHYLENE GLYCOL VAPOR DISTRIBUTION FOR AIR STERILIZATION†

By EDWARD BIGG*, M.D., CHICAGO, ILL., B. H. JENNINGS**, AND F. C. W. OLSON††, EVANSTON, ILL.

THE basic concept of air disinfection by the use of glycol vapors has now become well established and repeated laboratory observations have conclusively demonstrated its value.[1-3] Several published reports on field trials have shown that reduction in air-borne infection may be brought about when groups are exposed to bactericidal concentrations of propylene or triethylene glycol vapors.[4-6] It is obvious that further progress will be dependent upon the combined efforts of engineering and medical personnel; the former to develop and refine methods of introduction of vapor, insure effective distribution in the desired space, control desired concentrations, and gather physico-chemical data on the behavior of the vapor; the latter to correlate epidemiologic and bacteriologic data on the actual results obtained. The purpose of this paper is to describe additional apparatus and its application and report briefly the results of a controlled experiment at an army training camp.

The original plan of experiment was to treat one large classroom building housing approximately 5000 men; half of these men were to live in glycolized barracks and half in untreated barracks. Trainees in untreated classrooms and barracks were to constitute controls. The sharp reduction in inductees subsequent to the end of the war necessitated changing the program, and consequently it was possible to carry out statistical observations only on the value of glycol vapor in the sleeping quarters (barracks). However, installations had already been made in the classroom building and it was considered worth while to operate this equipment to determine its effectiveness and the feasibility of treating a building of this size, since no attempt had ever been made on such a scale.

---

†This work was done under contract between Northwestern University and the Office of Scientific Research and Development and the Office of the Surgeon General, U. S. Army.

*Associate in Medicine, Northwestern University Medical School.
**Professor of Mechanical Engineering, Northwestern Technological Institute. Member of A.S.H.V.E.
††Research Associate, Northwestern Technological Institute.

[1]Exponent numerals refer to Bibliography.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Coronado, Calif., June 1947.

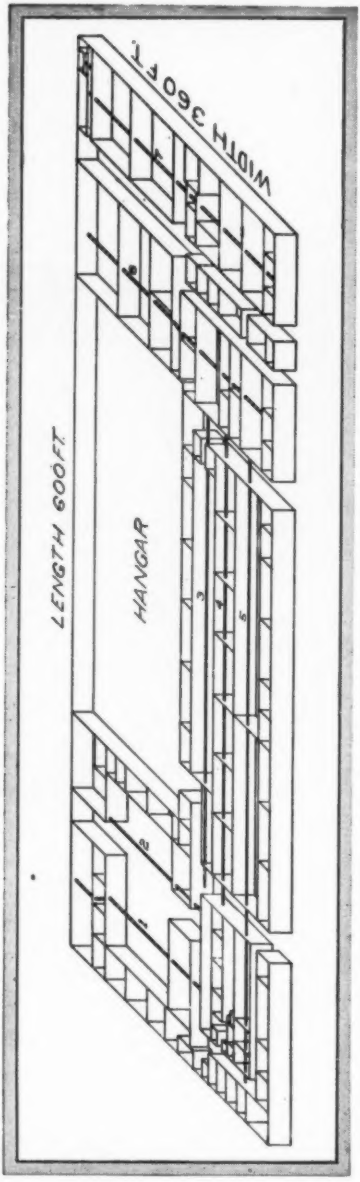


FIG. 1. DIAGRAMMATIC REPRESENTATION OF GLYCOL TREATED SCHOOL BUILDING SHOWING SEVEN DUCT RUNS.

DESCRIPTION OF EQUIPMENT

Building

This school building was 360 ft wide and 600 ft long, having a capacity of 3,500,000 cu ft, excluding the airplane hangar. The roof was of sawtooth construction. Fig. 1 shows a diagrammatic representation of this structure. The interior was divided by metal partitions into 105 rooms. A few of these partitions were completed, extending from the floor to the roof; most were partial, extending from the floor to a point about fourteen feet above the floor. The building was heated by steam; some of the smaller rooms had steam radiators, but most of the rooms depended upon unit heaters. It was obvious that in a building of this type a duct system would be necessary to obtain uniform distribution of glycol throughout the building. The duct layout is also shown in Fig. 1. As may be seen, seven lengths of duct were installed, each to treat a portion of the building, and each operating as an individual unit. Such a unit consisted of the duct, a vaporizer and a high pressure blower fan. The duct handled only recirculated air. Ventilation was obtained by opening windows and roof vents.

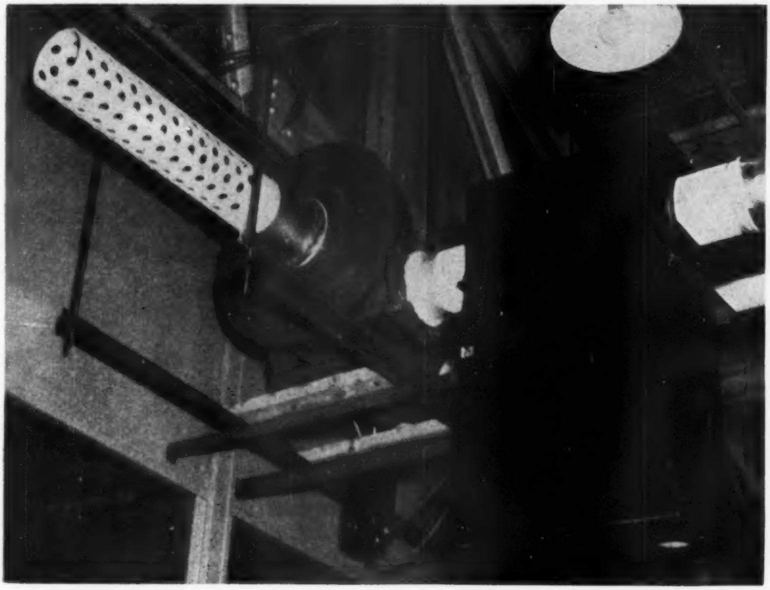


FIG. 2. BLOWER FAN AND GLYCOL VAPORIZER CONNECTED TO DUCT WITH INLET SILENCER IN PLACE.

Ducts

The ducts were constructed of three foot lengths of laminated asbestos. The sections were joined with asbestos collars cemented to the ducts. Uniformly spaced (9 ft) venturi-type openings for distribution were located on the lateral surface. Holes were cut through the metal partitions and the ducts were hung by cables attached to sheet metal collars and connected to overhanging guide

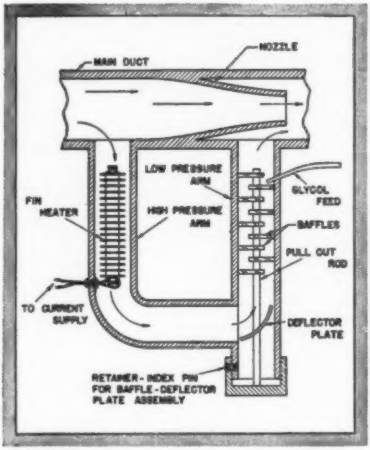


FIG. 3. DIAGRAMMATIC REPRESENTATION OF VAPORIZER ASPIRATOR USED FOR GLYCOL GENERATION.

wires. Each duct was sealed at the far end and since the ducts were from 300 to 460 ft in length, it was desirable to use six inch diameter ducts in order to reduce the pressure drop through the system. A total of 2300 ft of duct was installed.

Fans

The blowers used were of a radial-blade type delivering 800 cfm of air against eight inches of water pressure. The blower was driven by a three horsepower, 220 volt, three phase motor running at 3000 rpm. This high speed caused considerable noise, which was greatly reduced by the installation of a muffler in the air intake. This muffler, shown in Fig. 2, consisted of a three foot length of six inch diameter duct. Forty-eight one inch holes were cut in the side of the duct, a heavy screen was fixed inside the duct six inches from the fan, and steel wool was packed loosely in the duct. The noise reduction by this method was sufficient in all cases where the blowers were installed in cor-

ridors. One blower was installed in a classroom and required a special soundproof box. This box was built of sheet metal and lined with one inch hair felt. Excellent silencing was obtained by this method.

The choice of these blowers was dictated by availability of equipment during the war. It is quite probable that a specially designed blower driven by a one horsepower motor or less would be sufficient. However, it was necessary to use

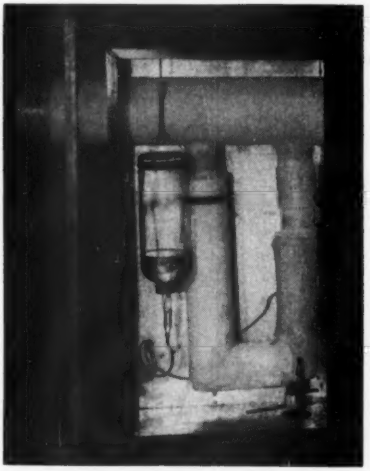


FIG. 4. PHOTOGRAPH OF VAPORIZER ASPIRATOR FOR GLYCOL GENERATION.

these oversize blowers because they and the motors were readily available and could be fitted to the duct system without special adapters.

Vaporizers

The problem of generating sufficient quantities of vapor and introducing it into the comparatively high pressure air in the ducts was solved by the construction of a *vaporizer-aspirator* shown diagrammatically in Fig. 3 and by photograph in Fig. 4. A constriction was placed in the main duct close to the blower. On the high pressure side, a take off arm permitted air to pass over a strip heater. The heated air then struck a deflector plate and passed upward over a series of baffles. A glycol-water mixture was fed on an upper baffle and then dripped down on successive baffles. The hot air was therefore in contact with a large wetted glycol surface on which vaporization took place. The air-vapor mixture entered the distributing duct in the low pressure area on the other side of the constriction. Insulation was required around the entire unit.

To permit inspection and cleaning of the baffles (which, however, proved to be unnecessary), they were mounted on a pull out rod so that the entire assembly could be withdrawn from the lower part of the unit. Feeding of glycol liquid through a needle valve was satisfactory, although it did require periodic inspection and adjustment.

When heating pure glycol, a fire hazard is present.[7] This can be practically eliminated by diluting the glycol with 5 to 10 percent water which was done.

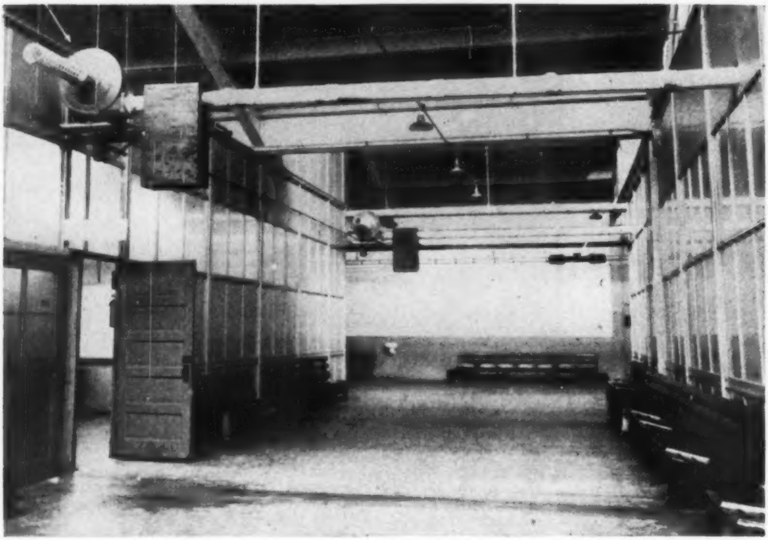


FIG. 5. GLYCOL VAPOR DUCT INSTALLED WITH FANS AND VAPORIZERS

The fin heater was so connected that in the event of blower failure the heater was automatically disconnected.

The vaporizer described was so devised that the quantity of glycol in it at any time was relatively small and remained in the vaporizer a comparatively short time. Therefore, the problem of deterioration of glycol on prolonged heating was practically eliminated. The output of the vaporizer could be closely controlled by varying the feed rate.

Operation of the System

In Fig. 5 is shown the fan with its inlet silencer drawing air from the room and delivering it into the duct system. Close to the start of the duct system can be seen the housing of the high capacity glycol vaporizer, from which

glycolized air passed into the duct system, where openings at appropriate intervals controlled the distribution into treated spaces.

Fig. 6 shows a duct in a classroom. The outlets are small molded ceramic fittings placed on approximately nine foot centers, and these delivered the glycol vapor into the upper portions of the room from which the glycolized air was distributed relatively uniformly to all portions of the room.

FIG. 6. LARGE CLASSROOM SHOWING OVERHEAD GLYCOL VAPOR DISTRIBUTION DUCT.

The concept of the use of a long duct with uniformly spaced small openings for the distribution of air and vapor at a relatively high static pressure is new. It was at first feared by some that most of the air would leave the first few openings and that at the far end of the duct the output would be small, and it was thought that a damper might be required for each opening to regulate the air flow. When the system was finally installed and operating, determinations were

TABLE 1—OUTPUT OF DUCT SYSTEM

DUCT NO.	STATIC PRESSURE AT ENTRANCE, IN. H_2O	RATIO OF OUTPUT OF LAST OPENING TO OUTPUT OF FIRST OPENING
1	7.0	0.96
2	8.0	0.94
3	1.7	1.00
4	1.8	0.94

made to demonstrate the uniformity of air flow and showed that the output from each opening was practically constant. Table 1 indicates the performance of the ducts at static pressures from 1.7 in. water to 8.0 in.

The glycol vaporizers used in the school building functioned satisfactorily for the purposes of the study, but certain modifications will be required before such equipment is adaptable for routine application. These would include a positive

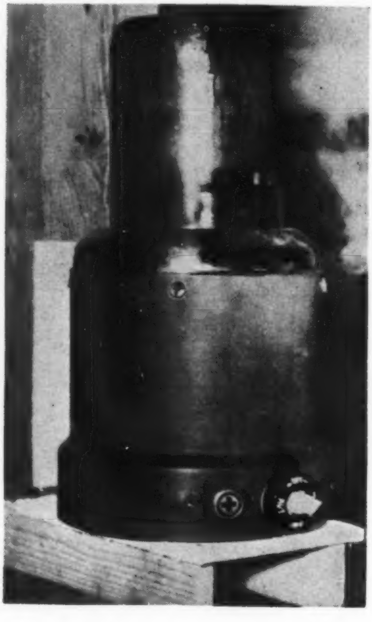


FIG. 7. INDIVIDUAL TYPE VAPORIZER USED WHERE CENTRAL SYSTEM WAS NOT INSTALLED.

feed mechanism, alteration in type of baffles, provision for rapidly adjusting and measuring air flow, and increasing the heat input so that a greater quantity of glycol may be vaporized when required. The blower used was larger than required; a smaller blower would reduce the noise substantially and with a lesser static pressure would reduce possibility for droplet carryover.

Glycol concentrations were determined daily when the equipment was in operation. With a glycol output per unit of approximately 80 cc per hour, the concentration was maintained between 3 and 5 micrograms triethylene glycol per liter of air in the space treated by that unit. There was a noticeable variation in the concentration from day to day which is best explained by the effect of

weather changes and the type of building construction. The openings in the duct were so placed that the glycol vapor was emitted in a horizontal stream. If the roof temperature and other conditions were such as to create an updraft, some of the glycol carried first to the ceiling and escaped through the roof windows before it could be recirculated toward the floor. On the other hand, with a prevailing downdraft, the glycol concentration quickly built up at the floor level. This variation could probably be reduced by placing the duct openings to blow downward at an angle of approximately 45 degrees.

Treatment of Barracks

It has been mentioned before that a duct distribution system was employed exclusively in the large school building. Because of the cost factor and other problems involved, it was decided to use individual glycol units in the barracks where the men resided. Because of the necessity of controlling a large number of individual units, these were not as satisfactory as the central system, but did deliver glycol into these spaces. The models available at that time have been considerably improved and modified, but an idea of their arrangement and operation should be of interest.

Fig. 7 shows one of the units mounted on a post in a barracks. These were operated by electrical heating of a glycol water solution, and a glycol water vapor passed into the barracks space. The vapor can be seen emerging from the outlets of the unit, and this visible vapor very quickly diffused into the atmosphere of the room and is not noticeable. These units were filled manually every two or three days. A more detailed description of the units is given in a previously published paper.[8] The control of glycol concentration in the barracks was necessarily crude. The units had three heat inputs which could be varied to give large changes in concentration. Adjustment between these ranges was made by altering the glycol concentration in the vaporizer feed bottle in the manner described.[8] Admittedly, this type of control is more an art than a science, but as found by experience, the average service man learned it well enough in a week or two to be able to maintain satisfactory concentrations in the barracks.

OBSERVATIONS ON DISEASE INCIDENCE

This study was limited to two squadrons, one of which was exposed to glycolized air in sleeping quarters (barracks). The mean strength of each squadron was approximately 500 men. Every effort was expended to obtain complete comparability of test and control groups since this essentially is the crux of such an experiment. Students were assigned to squadrons for purposes of housing and administration only, and were indiscriminately intermingled in classrooms, mess halls, and recreational activities. Each squadron, within the limits of chance, was composed of personnel with equal distribution as to age, length of military service, crowding in barracks, population turnover, and geographical origin.

Clinical data were based on cases hospitalized for those diseases believed to be disseminated by the aerial route and consisted of the diagnoses placed in the hospital record of patients at the time of discharge, at which time the diagnosis

was definitely established. Records included measles, mumps, *common cold,* atypical (virus) pneumonia, bacterial pneumonia, streptococcal infections of the respiratory tract, rheumatic fever, scarlet fever, etc. All cases of respiratory infections of bacterial nature and those of unknown etiology such as the common respiratory diseases hospitalized less than 10 days after arrival to the camp were omitted. This step was taken to eliminate those cases in which the incubation period is thought to be at least ten days and those in which an infection might have been contracted prior to coming under test observation. Similarly, the specific virus infections, measles, mumps and atypical pneumonia, were excluded in those cases where hospitalization occurred within 21 days. These periods are arbitrary but were applied equally to all men under study, both in test and control group. Disease rates were computed on the basis of rate per 1000 per annum.

A pretest observation period was begun December 1, 1945, and continued for eight weeks. The test period extended for 10 weeks, beginning January 25, 1946, and ending April 5, 1946. During the pretest period a disease rate of 262 per 1000 per annum occurred in the control group and 272 per 1000 per annum in the test group; during the period of *glycolization,* these rates were 714 and 384, respectively. This represents a 46.2 percent reduction in disease incidence brought about by the introduction of the glycol vapor into the barracks of the test group.

The medical and epidemiologic aspects of the foregoing statistics are to be elaborated and discussed in detail in a forthcoming publication.

CONCLUSIONS

1. The use of an independent duct system carrying glycol vapor and sufficient air for dilution appears the most effective method for distributing glycol vapor throughout large spaces.

2. For small spaces, unit type vaporizers in adequate number and properly located can maintain desired bactericidal concentrations.

3. Hospitalization due to respiratory infection was significantly reduced in groups sleeping in glycol treated quarters, confirming previous results of the authors.

ACKNOWLEDGMENTS

The help and advice of E. DeCamp of the Philip Carey Co. in the design and construction of the duct systems is greatly appreciated. In the conduct of the tests and supervision of the military and civilian laboratory personnel the assistance of Major Fred C. Garlock, Sn. C., was invaluable. This investigation was made possible through the whole-hearted cooperation of the Commanding Officers of the Army Air Forces, the staff at Chanute Field and the sponsorship of the Office of the Surgeon General, U. S. Army.

BIBLIOGRAPHY

1. The Use of Glycol Vapors for Bacterial Control in Large Spaces, by Edward Bigg, B. H. Jennings and S. Fried. (*American Journal Medical Science,* Vol. 207, p. 361, March, 1944.)

2. The Harvey Lecture Series, by O. H. Robertson, Vol. XXXVIII, p. 227, 1942-43.

3. The Fungicidal Action of Triethylene Glycol, by Margaret Mellody and Edward Bigg. (*Journal Infectious Diseases,* Vol. 79, p. 45, Aug., 1946.)

4. Summary of a Three Year Study of the Clinical Application of the Disinfection of Air by Glycol Vapors, by T. N. Harris and J. Stokes. (*American Journal Medical Science,* Vol. 209, p. 152, Feb., 1945.)

5. Epidemiologic Observations on the Use of Glycol Vapors for Air Sterilization, by Edward Bigg, B. H. Jennings and F. C. W. Olson. (*American Journal Public Health,* Vol. 35, p. 788, Aug., 1945.)

6. The Use of Glycol Vapors for Air Sterilization and the Control of Air Borne Infections, by B. H. Jennings, Edward Bigg and F. C. W. Olson. (A.S.H.V.E. TRANSACTIONS, Vol. 50, p. 343, 1944.)

7. Inflammability Characteristics of Propylene Glycol and Triethylene Glycol in Liquid and Vapor Form, by Edward Bigg, B. H. Jennings and S. Fried. (*Americtn Journal Medical Science,* Vol. 207, p. 370, March, 1944.)

8. Portable Glycol Vaporizers for Air Sterilization, by Edward Bigg, F. C. W. Olson and B. H. Jennings. (*Science,* Vol. 105, p. 23, Jan. 3, 1947.)

DISCUSSION

L. J. BUTTOLPH, Cleveland, Ohio (WRITTEN): The effectiveness of this installation in which glycol vapor was introduced relatively near the ceilings and from rather widely spaced fittings to secure a good distribution of vapor throughout the whole volume of the rooms, with a relatively small volume of air and air movement used to introduce the vapor, and with no mechanical air circulators in the rooms, seems to confirm observations of those who have studied the effectiveness of upper air ultraviolet irradiation in reducing the bacterial content throughout the lower air of a room. In both cases the practical results seem hard to interpret other than on the basis of a random air circulation throughout the whole room, considerably greater than is ordinarily visualized by ventilating engineers.

It would be interesting, if possible, to determine whether glycol vapor is distributed and maintained throughout the whole volume of such experimental rooms at concentrations continually effective *at the time and place of contamination,* or whether the germicidal action takes place largely in the upper air where the glycol concentrations are presumably considerably greater, with a lower air effect resulting from air circulation and dilution.

Although the opinion does not appear in this paper, advocates of both glycol and of ultraviolet air disinfection have sometimes maintained that rates of air decontamination sufficiently rapid to reduce the spread of respiratory diseases are impractical of attainment by ventilating dilution with fresh air. How practical natural ventilation may be for this purpose is obviously a function of the cost of heating and moving the required air. Perhaps this aspect of the problem of air disinfection or decontamination should be examined more closely.

Innumerable additional excellent studies such as this of the epidemiological value of air disinfection will probably be needed before there can be any final appraisal of its value. In the meantime the value of individual studies could be increased very greatly if there were some common criterion of air disinfection such as either the percent or the absolute reduction in index organisms, to correlate the epidemiological results secured by the three obvious methods of air disinfection. To what extent may

we assume that the epidemiological results of air disinfection are independent of the method of disinfection?

W. T. ANDERSON, JR., Newark, N. J. (WRITTEN): The Commission on Air-borne Infections, Army Epidemiological Board, *American Journal of Public Health,* Dust and Its Controls as a Means of Disinfection of Air, Vol. 37, p. 353, 1947, has reported that dust in army barracks becomes highly contaminated with certain pathogens associated with certain diseases of the respiratory tract, particularly those which may cause streptococci infections. These micro-organisms are dispersed into the air from bedclothes, floors, and clothing at times of floor sweeping, bedmaking, and dressing, and provide opportunities for the spread of disease by direct or indirect contact with the infected dust. Oiling of floors, bedclothes, and other textiles proved highly effective procedure for the control of dust-borne bacteria. The action is only a mechanical one. Studies have indicated that this oiling process can result in a very significant reduction in infections of streptococci origin. They concluded that its combination with glycol vapors, or ultraviolet, or adequate ventilation offered the best opportunity to control air-borne bacteria.

It appears unfortunate to the writer that in the present experiments with glycol vapors some attention was not paid to the effect of oiling of floors. If both the floors of the control and the glycol treated rooms had been oiled, would the differences in reduction of infection between the two groups have been so great? How much of the effect was due to glycol on the floor acting as an oil to hold the dust? How much was due to the glycol actually removing air-borne bacteria? In other words, can nearly the same results be obtained if the floors are swabbed with glycol liquid?

It is quite possible that the authors have the answers to my questions. If they do, I would appreciate greatly having literature references if the answers have been published. Unfortunately I do not have the answers, but the report of the Commission cited in the first paragraph is suggestive.

Attention is also directed to a paper entitled Recent Studies on Disinfection of Air in Military Establishments, *American Journal of Public Health,* Vol. 37, p. 189, February, 1947, which states that *The use of triethylene glycol must be supplemented with dust control measures if it is to be successful.*

W. F. WELLS, Philadelphia, Pa. (WRITTEN): By recognizing the significance of engineering design and operating efficiently in the environmental control of air-borne contagion, the authors have taken an important step in the development of chemical disinfection of air. A recent report from the *American Public Health Association* grants that air disinfection is indicated under particular conditions in hospitals, but pronounces general application at the present time and indiscriminate use at any time as unjustifiable. The ventilating engineer and the sanitarian are therefore charged with the responsibility of specifying the conditions and circumstances of sanitary ventilation.

An A.S.H.V.E. Technical Advisory Committee on Air Sterilization has been appointed by the Society to study the problem of sanitary ventilation and to present recommendations for a new chapter in THE GUIDE on this new ventilating art. The paper here presented will help much in defining the field of application of glycol vapors in air disinfection.

Perhaps some means of testing the efficiency of installations is the most pressing need of the ventilating engineer and the public health official. We atomized constant numbers of standard test organisms into ventilated spaces and determined by the air centrifuge the equilibrium concentration with and without air disinfection. This gave a measure of the sanitary equivalent in terms of air changes which would remove equal numbers of pathogenic organisms tested in the laboratory.

The isolation and identification of organisms contributed by occupants gives another method of measuring ventilation load; streptococci of the nose and throat can usually be isolated during the winter from five cubic foot samples of occupied atmospheres.

Ultimately these values should, as the authors say, be correlated with epidemiologic experience, though it may take as many years as it took to establish quality standards of drinking water supplies.

H. C. MURPHY, Berkeley, Calif. (WRITTEN): I have been much interested in the very excellent paper presented by Professor Jennings and would like to submit my comments. In Fig. 2 the authors show an *inlet silencer* and then state:

This muffler, shown in Fig. 2, consisted of a three foot length of six inch diameter duct. Forty-eight one inch holes were cut in the side of the duct; a heavy screen was fixed inside the duct six inches from the fan; and steel wool was packed loosely in the duct. The noise reduction by this method was sufficient in all cases where the blowers were installed in corridors. One blower was installed in a classroom and required a special soundproof box. This box was built of sheet metal and lined with one inch hair felt. Excellent silencing was obtained by this method.

In effect I believe this muffler is a fairly efficient air filter of the viscous impingement type. The loosely packed steel wool with the glycol coating picked up from the recirculated air would, I believe, have a measurable efficiency in removing air-borne organisms.

Dalla Valle and Hollaender of the *U. S. Public Health Service* indicated in Vol. 54, No. 17 of the *U. S. Public Health Reports* dated April 28, 1939—that steel wool filters three inches thick were found in their investigations to have efficiencies of 83 to 85 percent in removing a test organism (the B. subtiles) from an air stream.

Lemon, Wise and Hamburger in the August 1944 issue of *War Medicine,* Vol. 6, pp. 92-101 discuss their studies of the bacterial content of air in Army barracks. Their investigations covered the use of viscous impingement filters 2 in. thick and their studies established efficiencies as high as 94 percent in some cases for these units in the removal of air-borne bacteria. In their conclusions they state:

The chief factor in the reduction of the bacterial content of air passing through the heating unit is the air filtration system. Dry or oiled wire mesh filters seem to possess about equal efficiency in removing air-borne bacteria. Filters partially clogged with dust and lint are somewhat more effective than clean ones, but if too dirty they interfere with adequate air circulation. There is some evidence that settling out of bacterial particles in the plenum chamber of the furnace further reduced the number of air-borne bacteria. Neither furnace heat nor the centrifugal action of the blower appears to have any influence on the number of recirculated bacteria.

W. L. HOLLADAY, Los Angeles, Calif. (WRITTEN): In the controlled test of sleeping quarters, was any effort made to simulate a test in the untreated area? For example, were dummy vaporizers holding plain water installed in the untreated zone, so the two zones would look alike to the occupants, and the psychological factors in each zone would be equalized? If such a precaution were not exercised, there may be a possibility that some of the cases from the untreated zone were psychosomatic in character.

AUTHORS' CLOSURE: The authors appreciate the comments which have been made in the written discussions and in those which were orally presented at the meeting.

We are very much in agreement with Dr. Wells that some method of measuring the effectiveness of air disinfection be prepared. These standards would be of great help in making comparative analyses of the effectiveness of different methods of air purification control. The measure of *sanitary equivalent* which we used was a chemical analysis of the air to determine the amount of glycol present. Laboratory standards for effective bactericidal concentrations of glycol have been well established and our standard for effectiveness was based on these data. It was our feeling that since these concentrations were reached, we were obtaining effective bactericidal levels.

In reply to Dr. Anderson's comment regarding oiling, it should be mentioned that blankets and floors in the barracks were treated according to the standards established by the Commission on Air-Borne Infections. This procedure was applied in both test and control spaces. From a study of the literature available at this time, it would appear that the effectiveness of dust control measures alone in controlling air-borne disease has not been established. Our results confirm this observation.

Mr. Holladay questioned the psychological factors present in our test. We would like to stress the fact that our disease incidence was based only upon those cases admitted to the hospital for diagnoses and treatment. The severity of these infections and the bacteriologic evidence collected was such as to place them beyond the possibility of being psychosomatic in nature.

Mr. Buttolph brings out the same point as Dr. Wells does in regard to the need for standards of measuring the effectiveness of air disinfection. Mr. Buttolph's point as to where, in a given space, the disinfection takes place, is also quite pertinent. Our measurements for glycol concentration in the treated space were made at the lower level in the room in the breathing zone, and previous observations have shown that the concentrations indicated are bactericidal. It is quite possible that conditions could exist in a space where, through lack of diffusion or dissemination, parts of the room had glycol in sufficient concentrations to be bactericidal.

In reply to Mr. Murphy, it should first be mentioned that the epidemiological statistical data mentioned in this paper were obtained in the barracks and not in the space where the muffler and duct system were employed. Our paper in this connection demonstrated that glycol could be distributed through a space by means of an individual duct system. There is certainly no question that filters could remove from a system a large portion of the bacteria suspended on large particles.

In Memoriam 1947

Name	Joined
Howard C. Baker, Reading, Pa.	1925
Harry G. Black, Philadelphia, Pa.	1917
Moritz I. Blumenthal, Los Angeles, Calif.	1936
Harry A. Brinker, Kalamazoo, Mich.	1934
Thomas F. Campbell, Pittsburgh, Pa.	1928
Robert Close, Leonia, N. J.	1938
George I. Cornwall (*Life Member*) Elizabeth, N. J.	1919
Howard S. Denham, Malden, Mass.	1939
John Devlin, New Orleans, La.	1940
Francis C. Dorsey, Baltimore, Md.	1920
William A. Ebert, San Antonio, Tex.	1920
Nicholas P. Fenner, Elmhurst, Ill.	1927
John L. Foley, Cleveland, Ohio	1938
Samuel L. Goodwin (*Life Member*) Hasbrouck Heights, N. J.	1924
Roy F. Hahn,* Atlanta, Ga.	1936
Howard S. Hamilton, Milwaukee, Wis.	1940
Ernest C. Henry, Bay City, Mich.	1938
Jared A. Hill,* San Francisco, Calif.	1938
William Beach Hodge, Charlotte, N. C.	1934
Louis F. Hudepohl, Cincinnati, Ohio	1936
Charles W. Jenkins, Washington, D. C.	1945
William T. Jones (*Presidential Member*) Boston, Mass.	1915
Thomas Proctor Mandell, Boston, Mass.	1937
Henry Mathis, Chicago, Ill.	1921
Earle W. McMullen, Joplin, Mo.	1938
Robert C. Morgan (*Life Member*) Philadelphia, Pa.	1915
Clarence H. Mosher, Buffalo, N. Y.	1919
Clark H. Parkes, Dodge City, Kan.	1944
Walter V. Reynolds, New York, N. Y.	1928
Joseph E. Robb, Lawrence, Kan.	1936
Paul F. Schlick, St. Paul, Minn.	1940
Andrew Sheret (*Life Member*) Victoria, B. C.	1925
Chauncey A. Simonds, Grand Rapids, Mich.	1944
Theodore F. Steinhorst, Utica, N. Y.	1919
Webster Tallmadge, East Orange, N. J.	1924
Nelson S. Thompson (*Life Member*) Washington, D. C.	1897
J. Herbert Walker, Birmingham, Mich.	1915
James B. Way, Montreal, Que., Canada	1945
A. Gordon Wheler, Syracuse, N. Y.	1945
Corbett F. Whitton, Toronto, Ont., Canada	1945
Ray S. M. Wilde, Detroit, Mich.	1916
Claude L. Winter, Grand Rapids, Mich.	1945
J. W. Winterbottom (*Life Member*) Waterloo, Ia.	1914

* Died in 1946 but Society notified in 1947.

INDEX

A

B

C

D

E

F

G

H

M

N

V

W

Y

Z